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Office of Conservation and Solar Energy

Test Procedures for Central Air Conditioners, Including Heat Pumps

DEPARTMENT OF ENERGY

Office of Conservation and Solar Energy

10 CFR Part 430

Energy Conservation Program for Consumer Products; Test Procedures for Central Air Conditioners, Including Heat Pumps

AGENCY: Department of Energy.

ACTION: Final rule.

SUMMARY: The Department of Energy amends its test procedures for central air conditioners to include test procedures for heat pumps, and to modify the existing test procedures for other kinds of central air conditioners. These test procedures are part of the energy conservation program for consumer products established pursuant to the Energy Policy and Conservation Act. Among other program elements, the legislation requires that standard methods of testing be prescribed for covered products.

EFFECTIVE DATE: January 17, 1980.

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SUPPLEMENTARY INFORMATION:

A. Background

On October 1, 1977, the Department of Energy (DOE) assumed the authority of the Federal Energy Administration (FEA) for the energy conservation program for consumer products, pursuant to Section 301 of the Department of Energy Organization Act (DOE Act) (Pub. L. 95–91). The energy conservation program for consumer products was established by FEA pursuant to Title III, Part B of the Energy Policy and Conservation Act (Act) (Pub. L. 94–163).[1] Section 323 (42 U.S.C. 6293) of the Act requires that standard methods of testing be prescribed for covered products. Test procedures appear at 10 CFR Part 430, Subpart B.

Test procedures for central air conditioners were originally proposed by notice issued June 7, 1977 (42 FR 30401, June 14, 1977) and a public hearing on the proposed test procedures was held on August 4, 1977. Final test procedures for central air conditioners were prescribed on November 21, 1977 (42 FR 60150, November 25, 1977). DOE proposed to amend the test procedures on April 6, 1979 (44 FR 23468, April 19, 1979) in order to include testing procedures for heat pumps, and solicited public comments on the need to modify any provisions of the existing test procedures for other kinds of central air conditioners. A public hearing was held on June 13, 1979, at which the proposed changes were discussed.

B. Discussion of Comments

Comments were received from manufacturers, a trade association, and interested individuals. The major issues raised by the comments are discussed below.

1. *Central Air Conditioner Definition.* One commenter requested clarification of the applicability of the test procedures to packaged combination units. A packaged combination unit is a central air conditioner and a furnace in a single cabinet. Although this type of equipment is generally used in commercial applications, either the heating capacity or the cooling capacity of a given model could fall within the capacity limits set forth in the central air conditioner or furnace definitions in 10 CFR 430.2. If both heating and cooling capacities of a packaged combination unit fall within the defined limits, the unit is accurately characterized as a residential unit and the DOE test procedures are applicable to both the heating and cooling functions. If either of the functions, however, has a capacity greater than the defined residential limits specified in section 430.2, the unit is most likely to be used in a commercial application and is not covered by the test procedures in either the heating or cooling mode. DOE has modified the definition of central air conditioner in section 430.2 to clarify the treatment of these units in today's final rule.

2. *Energy Efficiency Measures.* Several comments were concerned with the use of the measures of efficiency identified in the proposed rule. Proposed section 430.22(m)(5) identified the heating seasonal performance factor (HSPF), the cooling seasonal performance factor (CSPF), the seasonal energy efficiency ratio (SEER), and the annual performance factor (APF) as measures of efficiency. The CSPF and the SEER are essentially the same; the CSPF is derived by dividing the SEER (expressed in Btu's/watt-hour) by a conversion factor (3.413 Btu's/watt-hour) to produce a dimensionless efficiency measure. Commenters argued against the use of the CSPF on the basis of present consumer familiarity with the SEER concept, and the limited, less comprehensible range of numerical values of the CSPF compared to the SEER. A related comment cautioned that heating efficiency and cooling efficiency should be expressed in the same units so that they can be easily compared.

The Federal Trade Commission (FTC) has requested that DOE provide the SEER as a measure of efficiency for use in the FTC appliance labeling program established by section 324 of the Act. DOE has reviewed all comments concerning energy efficiency measures and has made the following changes from the proposed rule:

1. Deletion of the CSPF from section 430.22(m) (5) and from the test methodology in Appendix M.

2. Retention of the SEER in section 430.22(m)(5) and section 5.1 of Appendix M as the only measure of cooling seasonal efficiency.

3. Alteration of the method of the calculation of the HSPF values measured in section 5.2 of Appendix M to reflect the multiplication of the conversion factor of 3.413 Btu's per watt-hour to result in the same units as the SEER.

4. Revision of the APF calculations in section 5.3 of Appendix M to replace the CSPF values with the SEER values derived in section 5.1.

Another commenter suggested that DOE prescribe only a national HSPF for comparison purposes, and delete the proposed regional HSPF values. The regional values discussed in the proposed rule are based upon climatic variations among specified geographic regions. DOE maintains that due to wide climatic variations in the United States, a single average HSPF value would not provide sufficient useful information to consumers and could be misleading when compared to other heating systems whose efficiency, unlike heat pump efficiency, is not affected by climate. Consequently, DOE has retained the regional HSPF values in section 430.22(m)(5) of today's final rule.

A final related comment pointed out potential problems concerning representations of product efficiency based upon computations indentified in Appendix M but not identified as "measures of efficiency" in section 430.22(M)(5). DOE intends only those measures specifically identified in section 430.22 to be used in representations regarding the energy consumption of covered products, but regulation of such representations is a

[1] Subsequently, the Act was amended by the National Energy Conservation Policy Act (NECPA) (Pub. L. 95–619). References in this notice to "the Act" or to sections of the Act refer to the Energy Policy and Conservation Act as amended by NECPA.

matter relegated to the Federal Trade Commission (FTC) by sections 323(c) and 333(c) of the Act.

3. *Degradation Coefficient.* Determination of the degration coefficient (C_D), used to quantify the cycling performance of central air conditioners and heat pumps, is necessary in order to evaluate seasonal performance. Section 2.1 of proposed Appendix M allowed manufacturers to test (by performing Tests C and D) to determine C_D for all basic models that they sell or, at manufacturers' option, to use an assigned value of C_D equal to 0.25. Many comments were received regarding the degradation coefficient. These comments pointed out two general areas of concern. The first concern was the proposed requirement that any manufacturer electing to use the assigned value to rate any model must use the assigned value to rate all models that the manufacturer produces. The second concern was the specified numerical value of the assigned degradation coefficient, 0.25.

Nearly all commenters urged that DOE allow actual testing of certain models in a product line without prohibiting the use of the assigned value on other models in that same product line. The commenters stated that the proposed requirement would be unduly burdensome since the time required to perform Tests C and D on all models would be considerable and would reduce the level of effort that a manufacturer could devote to developing and testing more efficient product designs. Commenters also stated that the testing cost for determining C_D would not be justified for low volume models and on older models which would soon be removed from production, but that sufficient marketing incentive exists to encourge such testing for newer, more efficient models with higher sales volumes. One commenter stated that his company had difficulty meeting the Tests C and D tolerances with certain units and that this difficulty, would, under the proposed rule, prevent him from advertising the higher ratings for other models which had been statisfactorily tested for C_D.

Some commenters stated that the proposed rule would favor the large manufacturers who had larger testing capabilities, but another commenter also objected to any rule which would allow manufacturers an option of testing certain models for C_D and using the assigned value on other models. This objection may have been based upon a concern that large mannufacturers could test all products in their product lines for C_D but could then calculate seasonal efficiency by using the assigned degradation factor of 0.25 for all products on which the measured C_D was greater than 0.25. DOE is aware that this situation could occur under today's final rule, as it could occur under the previous test procedures for central air conditioners which have been in effect since January 1, 1978, but DOE has found no evidence of its occurrence.

Based on its review of all comments, DOE has determined that manufacturers of central air conditioners and heat pumps should be given the option to choose which models to test for C_D, retaining the option to use the assigned C_D for all other models. In addition, in the case of heat pumps, the final rule clarifies that manufacturers will be allowed the option of testing a model for C_D on a unit in the cooling mode and then using the assigned C_D for that same unit in the heating mode, and vice versa. However, in the case of a unit with two compressor speeds, if the assigned degradation coefficient is used at one speed, it must also be used for the other speed. Sections 2.1, 2.2, and 3.1.3 of Appendix M have been rewritten to reflect these changes.

Many comments were also received expressing concern over the numerical value of the assigned degradation coefficient, 0.25. It should be noted that lower degradation coefficients indicate higher seasonal efficiency. Two commenters stated their belief that the 0.25 value accurately reflects the data presently available; two other commenters urged that the assigned C_D should be reduced to 0.15 to reflect lower cyclic losses based on data from a study conducted at Purdue University's Herrick Laboratories under the sponsorship of the Air Conditioning and Refrigeration Institute. Another commenter stated that the Herrick data presently available showed only that further investigation is necessary. Both DOE and the National Bureau of Standards (NBS) have carefully considered these comments and DOE has concluded that available data support the assigned 0.25 value for C_D, which is only slightly higher than the mean of the C_D values currently known to DOE which have been determined by actual testing. Units with good cycling performance can only be identified by testing to determine C_D. Setting the assigned value for C_D to be slightly higher than the average value will give manufacturers an incentive to test their units to show the actual cycling performance. At the same time, since the assigned value is close to the mean, manufacturers who do not test for cycling performance are unlikely to be severely penalized when they use the assigned value. Consequently, no changes in the assigned value of C_D have been made in today's final rule.

4. *Cyclic Test Procedures.* One commenter disagreed with the "on" and "off" times specified in Test D in section 3.1.1.2 of Appendix M for use in measuring the degradation coefficient. The commenter submitted data showing cycle times for actual field installed units in both the heating and cooling modes. The data were carefully analyzed by NBS and the results were compared with predictions made by a computerized thermostat model. NBS found that the field results provided insufficient data to conclude that heat pumps typically operate at longer on-times than was assumed in DOE cyclic test procedures. Based on a recommendation by NBS, today's final rule relating to cyclic "on" and "off" times remains the same as was originally proposed.

Another commenter suggested a new procedure termed "time constant evaluation test" to replace the cyclic tests for determination of cyclic degradation. The time constant approach is presently under study by NBS. While this method may have advantages such as simplified procedures and reduced test time, the tests which thus far have been conducted according to this method raise doubts about its ability to measure accurately the cyclic performance degradation for all the different types of air conditioners and heat pumps presently manufactured. The time constant approach to part load performance shows promise but is not yet sufficiently developed to the point where it can be utilized.

5. *Evaporator Coils.* The proposed rule required that within a family of basic models that use the same condensing unit, only the condenser-evaporator coil combination expected to have the largest volume of retail sales must be tested. Ratings for other basic models could, at manufacturers' option, be determined by computer simulation methods which would predict the performance of other basic models consisting of the same condensing unit but different evaporator coils. The proposed rule specified certain minimum criteria for these computer models and required that any manufacturer who elected to use computer simulation submit to DOE the simulation method used. DOE also solicited comments on the feasibility of DOE prescribing a computer model for use by all manufacturers who chose not to rate their products by actual testing.

Comments received from manufacturers who had computer models unanimously opposed the idea of sumitting these models to DOE because of their perceived proprietary character. Commenters also expressed doubt that a single DOE prescribed computer model could be specific enough to predict accurate ratings, yet flexible enough to accommodate all manufacturers' products and permit innovative designs. Many commenters requested that DOE also allow manufacturers to use graphic and manual engineering calculation procedures to predict performance of condenser-evaporator coil combinations not actually tested.

Manufacturers who produce only evaporator coils expressed concern that builders might be reluctant to install split system air conditioners consisting of a condensing unit made by one manufacturer and an evaporator coil made by a different manufacturer, unless the evaporator coil manufacturers could provide the builders with information concerning the energy characteristics of the installed systems. Because the evaporator coil manufacturers typically lack the facilities or resources for testing their coils with significant numbers of other manufacturers' condensing units, these commenters suggested that the DOE test procedure should apply to only the condensing unit of split system air conditioners, since it accounts for the largest portion of the energy used by the total system.

DOE has carefully considered all of these comments and has concluded that applying the test procedure to condensing units only would require the deletion of seasonal energy efficiency ratios from the test procedures. This would be detrimental to consumers, because seasonal energy efficiency ratios best approximate actual energy use in homes. On the other hand, DOE is also aware that any requirement that manufacturers test all condenser-evaporator coil combinations would be unduly burdensome. DOE has concluded that the use of a computer simulation technique is the most accurate method for establishing rated values for all basic models of split system central air conditioners and heat pumps which are not actually tested. Since graphic and manual engineering calculation procedures perform the same functions as do computer simulation techniques, DOE has determined that graphic and manual engineering calculation procedures are also acceptable for predicting the performance of models of split system central air conditioners which utilize different evaporator coils. The National Bureau of Standards (NBS) is currently developing a computer simulation technique to predict the performance of any evaporator coil with any condensing unit of known performance characteristics. If this simulation method is shown to be accurate, the program will be made available to all manufacturers for use at their discretion. Although the use of analysis techniques is retained as an option available to manufacturers, whenever measures of energy consumption determined by analysis do not agree with measures of energy consumption outlined by actual testing, the test results shall prevail.

Consequently section 430.23(m) of today's final rule permits ratings of condenser-evaporator coil combinations to be based upon the results of condenser manufacturers' tests of each condensing unit with the evaporator coil likely to have the largest sales volume of all coils that the manufacturer matches with that particular condensing unit. The manufacturer may then rate all other condenser-evaporator coil combinations using that particular condensing unit with values obtained by using computer simulation or other suitable analysis which incorporates the minimum requirements outlined in section 430.23(5) to predict the performance of other evaporator coils with that condensing unit.

DOE has investigated thoroughly the provision in the proposed rule that requires manufacturers to submit copies of their computer model to DOE if they use a computer simulation technique to rate condenser-evaporator coil combinations different from those that are actually tested. DOE has concluded that it would be difficult for the Government to examine and certify the accuracy of each individual manufacturer's simulation method. The accuracy of simulation methods used by a manufacturer can only be determined by verification testing of actual condenser-evaporator coil combinations and comparison with results predicted by the simulation technique. As further explained in part C.3 of the preamble, verification of the accuracy of measures of energy efficiency reported for labeling of consumer products according to today's final rule is the responsibility of the Federal Trade Commission (FTC). The FTC may decide to determine the accuracy of these representations by testing certain condenser-evaporator coil combinations that were rated by simulation methods. In such instances the accuracy of the manufacturer's simulation methods is of critical importance since it is anticipated that the accuracy of the representations will be judged in comparison with the actual test results.

However, DOE must retain some method of substantiating the accuracy of manufacturer's simulation methods in order to preserve the objectivity of the central air conditioner test procedure. Consequently, DOE has revised the final rule to include the requirement that manufacturers who choose to rate certain condenser-evaporator coil combinations of central air conditioners by the use of simulation methods must submit to DOE a written statement which describes the type of simulation method used. In addition, users of simulation methods must, for every family of basic models using the same condensing unit, submit to DOE the test results used to establish the rated value(s) of efficiency for the combination likely to have the highest sales volume, and a comparison of these test results with the system efficiency predicted when the simulation method is used to determine the efficiency of the system with the highest sales volume coil.

If the ongoing NBS efforts succeed in developing an accurate computer simulation method for predicting the performance of condenser-evaporator coil combinations, DOE may use this technique as an initial screening method to determine which manufacturers' simulation methods shall be more closely inspected. Section 430.23(m)(8) has been added to require any manufacturer which elects to rate its products by simulation methods to permit representatives of DOE to inspect those methods to verify their accuracy. DOE anticipates that such inspections will be performed at the manufacturer's site and will involve analytical evaluation of the simulation methods used, as well as actual performance of simulations and comparison of predicted results with actual test results. Inspections at the manufacturer's site are expected to be the most convenient and efficient means of verifying the simulation methods used. DOE will retain the option of requiring manufacturers to submit their simulation methods to DOE pursuant to the authority of section 329 of the Act if this is determined in the future to be necessary, either in the case of a particular manufacturer or for all manufacturers which use the simulation option. DOE plans to inspect and verify by actual testing a significant percentage of manufacturer's simulation methods each year.

After thorough review of the comments received from manufacturers

who produce only evaporator coils, DOE acknowledges the need to provide these manufacturers with a method by which they can make representations of the efficiency of split system central air conditioners which use their evaporator coils in combination with other manufacturers' condensing units. DOE agrees that a rule which requires builders or coil manufacturers to test their evaporator coils with the condensing units of numerous other manufacturers would be burdensome. However, it is equally important that an accurate rating procedure for these split systems be established to ensure that consumers will be able to determine the estimated cost of operation and the efficiency of these combinations. DOE has concluded that an analytical method which utilizes computer simulation or other engineering analysis to predict the performance of these combinations is the most accurate alternative to testing. The method to predict the performance of the condenser-evaporator coil combination must be based upon condensing unit performance data provided by the manufacturer of the condensing unit. Today's final rule allows the use of this technique for evaporator coil manufacturers to rate split system combinations using their evaporator coils with another manufacturer's condensing unit. This approach will provide consumers and installers with accurate energy consumption and cost of operation information for these systems and will also protect evaporator coil manufacturers, which are typically small businesses, from any competitive disadvantage. It is anticipated that regardless of the method used to rate the product (testing, computer simulation, or engineering analysis), determination of compliance by the Federal Trade Commission or DOE may require actual testing to substantiate the measures of energy consumption published.

6. *Demand Defrost.* Many comments were received concerning both the definition of "demand defrost" in proposed section 1.12 of Appendix M and the application of an enhancement factor referenced in section 5.2 of Appendix M to show improved performance for units incorporating this feature. All comments concerning the definition stated that the wording was too prescriptive and would not allow innovative methods to be incorporated into production as new technology is developed. DOE has revised the definition of "demand defrost" in section 1.12 to cover all types of demand defrost which initiate the defrost cycle only by measurement of some variable parameter(s) in order to establish a need for defrosting, due to frost build-up on the outdoor coil.

One commenter stated that the energy consumption of demand defrost should be measured by actual testing rather than estimated by assigned factors. DOE is aware that testing is the ideal means by which to evaluate this feature; however, DOE has determined after consultation with NBS that considerably more research is needed before a methodology can be developed which adequately accounts for climatic variation. Because available information indicates that demand defrost systems do result in significant energy savings, DOE has determined for the present to evaluate demand defrost improvement by use of an enhancement factor derived from correlation to actual field test results.

Many commenters apparently attributed the proposed 5 percent enhancement factor specified in section 5.2 of Appendix M directly to seasonal performance improvement. In actuality the 5 percent factor applies to the defrost capacity (Q_{DEF}) in section 5.2 which in turn improves the heating seasonal performance factor (HSPF) by approximately 3 percent. Those comments which criticized the enhancement factor provided no data to indicate that demand defrost does not typically use less energy than time-temperature defrost over the course of a year. Some comments provided detailed analysis and field test results which agreed with the proposed approach and showed that demand defrost does improve the seasonal performance of a heat pump in the heating mode. Although this improvement does vary by region, analysis shows that in most areas a 4 percent improvement in the seasonal performance factor is the minimum which would be expected. Based on the information submitted and actual test results, DOE has adjusted the enhancement factor in section 5.2 of Appendix M to 7 percent, in order to achieve a resultant effect of 4 percent improvement in the HSPF for units which use demand defrost.

7. *Water Source Heat Pumps.* Two commenters stated that the test methodology proposed to be used for water source heat pumps was not consistent with that used for air source units. Commenters argued that these inconsistencies could give erroneous cost information and encourage sales of one product over another. After reviewing the comments associated with this issue, and considering the present low sales volume of water source heat pumps, DOE has determined not to include a methodology for testing water source heat pumps in today's final rule. However, it is expected that after further study by DOE and NBS a test methodology for these products will be proposed as soon as practicable. In today's final rule all sections which referred in the proposed rule to water source heat pumps have been reserved.

8. *Dual Voltage Units.* In section 3 of the proposed rule, DOE required testing of central air conditioners at only one voltage (either 115V or 230V), which was determined to be the typical residential voltage for a particular basic model. Some comments specifically requested that the test procedures permit the rating of central air conditioners at voltages other than 230 or 115 volts by correlative methods or computer simulation. Because voltages other than 230 or 115 are not commonly used in residential applications, DOE foresees no need for specifying test procedures at other voltages.

9. *Hours of Operation.* One commenter remarked that the heating and cooling load hour maps referred to in sections 6.1.3 and 6.2.5 of proposed Appendix M did not correlate with the actual hours of operation which the commenter expected to find in a particular city. The values on the maps were not designed to represent any particular city, but rather an average of the cities within a region. These average values reflect national variation in climate in sufficient detail for purposes of this program. Therefore, today's final rule reflects no changes in the heating and cooling load hours.

10. *Estimated Annual Heating Costs of Heat Pumps.* Several commenters expressed concern that the estimated annual heating costs for heat pumps in proposed sections 430.22(m)(1) and 430.22(m)(3) would be overstated by the proposed test procedure. DOE is aware that the sizing calculations made for furnaces and heat pumps typically employ many conservative, "worst case" assumptions. DOE proposed a correction factor of 0.77 in the sections mentioned above. The 0.77 value has traditionally been used by industry for estimating actual heating season costs more accurately. It is significant that this same correction factor is used in DOE's furnace test procedures, and its retention here facilitates comparisons between the operating costs of heat pumps and the operating costs of furnaces. While the actual costs may differ somewhat, the relative operating cost differences between furnaces and heat pumps are accurately developed by the consistent use of the 0.77 correction factor. Until further field test data are

available which establish the need for a change, the test procedures will continue to use the 0.77 correction factor.

11. *Sampling Plan.* In addition to the one-sided sampling plan contained in proposed section 430.23(m), the notice of proposed rulemaking also solicited comments on an alternate sampling plan in which testing would have been performed on only two units of each basic model required to be tested, with the mean value of the test results forming the basis for product ratings. Comments on these two sampling plans were divided, with some commenters favoring the proposed plan because of the more reliable results it would yield, and with other commenters favoring the alternate plan because it might reduce the number of units required to be tested. One manufacturer provided data which indicated that with four of the six models it had tested, testing of two units was sufficient to satisfy the confidence levels required in the one-sided sampling plan.

The Air Conditioning and Refrigeration Institute (ARI) suggested an alternate sampling plan to be used in conjunction with a future certification/enforcement program for labeling and efficiency standards. This plan would allow manufacturers to rate their products based upon a minimum of one test. Under this plan, ARI would modify an existing industry-supported certification and enforcement program so that verification of published ratings would be done by testing models selected under the following criteria:

(a) selection due to a "for cause" evaluation determined by trade association engineers,

(b) selection based on shipment volume, and

(c) arbitrary selection of existing models.

This verification plan as presently formulated does not utilize the statistical confidence levels proposed by DOE. Instead, a negative tolerance of 5 percent is allowed on capacity and SEER and if either of two units tested meets this tolerance, the product's rating is verified. If not, three additional units are tested and the product is then rated at the mean of the measured values of the last three units tested.

The alternate program offered by ARI combines testing and enforcement activities into a single program. The DOE enforcement program for use in conjunction with the forthcoming energy efficiency standards for air conditioning equipment is still under development. All comments made which pertain to enforcement and certification are being considered in the development of the enforcement program. DOE recognizes the merits of the existing industry certification programs and will consider incorporating any appropriate portions of those programs into DOE's own enforcement plan for the standards program.

DOE has considered all comments received relating to the two sampling plans offered for comment in the proposed rule. Since the data which were received show that the proposed plan utilizing statistical confidence levels will not require significantly more testing than the two-unit plan, DOE has retained the one-sided statistical sampling plan in today's final rule because of the increased confidence attributable to values of energy consumption measured under this plan.

12. *Industry Standards.* Many of the comments received indicated that two industry standards referenced in the proposed test procedure have subsequently been revised, and consequently commenters urged that the final test procedure reference the latest version of these standards. DOE has reviewed both editions of the American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) Standard 37 and the Air Conditioning and Refrigeration Institute (ARI) Standard 210. No substantive changes in any sections referenced in the DOE test procedure were found. For this reason, and because the latest versions of these standards can more easily be obtained by interested persons, today's final rule references the current versions, ASHRAE 37–78 and ARI 210–79. As noted below, in part C.1. of this preamble, DOE has consulted with the Attorney General and the Chairman of the FTC regarding DOE's use of these industry standards.

13. *Test Measurements and Tolerances.* Some comments stated that the dew point measurement specified in section 4.2.2.3 of proposed Appendix M was not consistent with wet bulb measurements in other parts of the test procedure. It was not the intent to require manufacturers to measure dew point, but simply to impose a tolerance of ±0.5°F on the resultant dew point determination when wet and dry bulb temperatures were required to be measured. DOE has revised Section 4.2.2.3 of Appendix M to clarify that dew point need not be measured.

Another comment requested a relaxation of tolerances on wet bulb temperature measurements from ±0.3°F to ±0.6°F. No specific reasons were given for the request nor was any substantive information provided. A careful examination revealed that ±0.3°F has been used in ASHRAE Standards for several years and therefore any changes to the proposed rule would not be consistent with past industry practice and would result in less reliable test results. In view of these facts, DOE has made no changes in the tolerances allowed for wet bulb measurements in today's final rule.

Other commenters recommended a more exact rounding procedure for capacity values specified in section 3.1 of Appendix M. The recommended rounding was 100 Btu/hour for capacities under 20,000 Btu/hour, 200 Btu/hour for capacities between 20,000 and 37,800 Btu/hour, and 500 Btu/hour for capacities between 38,000 and 64,500 Btu/hour. DOE agrees that this recommendation promotes greater accuracy without affecting test procedure costs, and therefore DOE has prescribed requirements in Sections 3.1 and 3.2 of Appendix M which allow a similar rounding off procedure in today's final rule.

14. *Miscellaneous.* After careful consideration of all the comments and further consultation with NBS and FTC, DOE has incorporated into the final rule some editorial and minor technical changes that were not discussed above.

C. Regulations Prescribed

1. *Test Procedures.* The test procedures for central air conditioners prescribed today are included in Subpart B of Part 430 and are substantially the same as those proposed, with the exception of the changes discussed above. Appendix M of Subpart B incorporates in part the revised Air Conditioning and Refrigeration Institutes Standards 210–79, 240–77, and 320–76, and the revised American Society of Heating, Refrigeration, and Air Conditioning Engineers Standard 37–78 to measure the capacities and efficiencies of central air conditioners, including heat pumps. Any subsequent amendment of these standards by the standard-setting organizations will not affect the DOE test procedures, which can only be amended by DOE.

Under the requirements of section 32(c) of the Federal Energy Administration Act of 1974 (15 U.S.C. 761 *et seq.*) as amended by section 9 of the Federal Energy Administration Authorization Act of 1977 (Pub. L. 95–70), DOE is directed to consult with the Attorney General and the Chairman of the Federal Trade Commission concerning the impact on competition of any rules prescribed by DOE which utilize or incorporate any commercial standards.

DOE has provided copies of the proposed test procedures for central air

conditioners, which incorporated the above-mentioned commercial standards, to the Attorney General and the Chairman of the Federal Trade Commission for their comments concerning the impact of such standards on competition. Neither official recommended against the incorporation or use of these commercial standards in the final test procedures for central air conditioners. At the request of the Attorney General, DOE will monitor manufacturers' costs of performing tests according to today's final rule.

2. *General Provisions.* Today's rulemaking contains definitions of "air-source heat pump," "central air conditioner," "condenser-evaporator coil combination," "condensing unit," "cooling only unit," "evaporator coil," "heat pump," and "water-source heat pump."

3. *Application of Test Procedures.* The test procedures prescribed today contain statistical sampling provisions for any testing required in order to comply with sections 323(c) and 324 of the Act. Section 323(c) provides that, effective 180 days after a test procedure rule applicable to a covered product is prescribed by DOE, no manufacturer, distributor, retailer, or private labeler may make any representation in writing or in any broadcast advertisement respecting the energy consumption or cost of energy consumed by that covered product, unless the product has been tested in accordance with the DOE test procedure, and the representation fairly discloses test results. Section 333(c) of the Act assigns enforcement responsibility for this provision to the Federal Trade Commission (FTC). The test procedures prescribed today are also applicable to any labeling rules which may be prescribed by the FTC pursuant to section 324 of the Act.

As required by section 325 of the Act, DOE is currently developing energy efficiency standards for the consumer products covered by the Act. An advance notice of proposed rulemaking relating to standards for nine types of consumer products, including central air conditioners other than heat pumps, was published in the Federal Register on January 2, 1979 (44 FR 49). An advance notice relating to standards for heat pumps is expected to be published in the near future, and notices of proposed rulemaking for standards will be published as soon as practicable.

The test procedure provisions prescribed today, other than the sampling provisions of section 430.23(m), are expected to be used generally for determining compliance with future standards, but the exact method of determining compliance will be decided during the ongoing standard setting process. The sampling provisions of section 430.23 are specifically intended to apply only to testing required pursuant to sections 323(c) and 324 of the Act. Future DOE regulations relating to the selection of units for testing to insure compliance with minimum efficiency standards under section 325 may differ substantially from the approach taken in the sampling provisions of section 430.23. One of the approaches that will be considered by the standards program for split-system central air conditioners would require testing of each condensing unit with the least efficient evaporator coil that is marketed with that particular condensing unit.

4. *Unit Costs of Energy.* Under section 323(b)(2) of the Act, DOE is required to provide manufacturers with information on representative average unit costs of energy. This information was provided by notice issued June 22, 1979 (44 FR 37534, June 27, 1979), and is expected to be updated periodically.

5. *Preemption.* Today's rulemaking prescribing final test procedures for central air conditioners supersedes any State regulation to the extent required by section 327 of the Act. Pursuant to section 327, all State regulations which provide for the disclosure of information with respect to any measure of energy consumption of central air conditioners, or which provide for an energy efficiency standard, or other requirement regarding the energy efficiency or energy use of central air conditioners, must now employ test procedures identical to those specified in today's final rule.

6. *Regulatory and Environmental Review.* In accordance with the requirements of the National Environmental Policy Act of 1969 (NEPA) (42 USC 4321 *et seq.*), DOE evaluated the proposed establishment of these test procedures for consumer products to determine if an environmental assessment (EA) or an environmental impact statement (EIS) was required. These test procedures will be used only to standardize the measurment of energy usage for the subject consumer products. The action of prescribing these test procedures, by itself, will not result in any environmental impact. Since it was thus clear that the proposed action was not a major Federal action significantly affecting the quality of the human environment, DOE determined that neither an EA nor an EIS was required. The potential environmental impacts that might occur from the application of the test procedures in connection with DOE's minimum efficiency standards program will be evaluated by that program.

This rule has been reviewed in accordance with Executive Order 12044 and DOE Order 2030. It was determined that this rule was significant in nature but did not have major impacts to manufacturers and consumers (imposing annual economic costs of $100 million or more).

(Energy Policy and Conservation Act, Pub. L. 94-163, as amended by Pub. L. 95-619; Department of Energy Organization Act, Pub. L. 95-91)

In consideration of the foregoing, Part 430 of Chapter II of Title 10, Code of Federal Regulations, is amended as set forth below, effective January 17, 1980.

Issued in Washington, D.C., December 10, 1979.

Maxine Savitz,

Acting Assistant Secretary, Conservation and Solar Energy.

1. Section 430.2 is amended by deleting the present definitions of "central air conditioner," "condenser-evaporator coil combination," "condensing unit" and adding the following new definitions in appropriate alphabetical order:

§ 430.2 Definitions.

* * * * *

"Air-source heat pump" means a "heat pump" which consists of one or more assemblies, which utilizes an indoor conditioning coil, compressor(s), and refrigerant-to-outdoor-air heat exchanger to provide air heating, and which may also provide air cooling, dehumidifying, circulating, and air cleaning.

* * * * *

"Central air conditioner" means a consumer product which is powered by single phase electric current, which is rated below 65,000 Btu/hour, which is not contained within the same cabinet as a furnace whose rated capacity is above 225,000 Btu/hour, and which is either a "heat pump" or a "cooling only unit".

* * * * *

"Condenser-evaporator coil combination" means a condensing unit made by one manufacturer and one of several evaporator coils, either manufactured by the same manufacturer or another manufacturer, intended to be combined with that particular condensing unit.

* * * * *

"Condensing unit" means a component of a central air conditioner which is designed to remove the heat absorbed by the refrigerant and to

transfer it to the outside environment, and which consists of an outdoor coil, compressor(s), and air moving device.

* * * * *

"Cooling only unit" means a "central air conditioner" which consists of an air cooled condensing unit and an evaporator coil, and which is designed to provide air cooling, dehumidifying, circulating, and air cleaning.

* * * * *

"Heat pump" means a "central air conditioner" which is either an "air-source heat pump" or a "water-source heat pump."

* * * * *

"Water-source heat pump" means a "heat pump" which consists of one assembly which utilizes an indoor conditioning coil with air moving means, compressor(s), and refrigerant-to-water heat exchanger(s) to provide both air heating and cooling, dehumidifying, circulating, and air cleaning.

2. Section 430.22 is amended by substituting a new paragraph (m) to read as follows:

§ 430.22 Test procedures for measures of energy consumption.

* * * * *

(m) Central Air Conditioners. (1) The estimated annual operating cost for cooling only units and air-source heat pumps shall be one of the following:

(i) For cooling only units or the cooling portion of the estimated annual operating cost for air-source heat pumps which provide both heating and cooling, the product of (A) the quotient of the cooling capacity, in Btu's per hour, determined from the steady-state wet-coil test (test A) measured at the highest compressor speed, as described in Section 3.1 of Appendix M to this subpart, and the average seasonal energy efficiency ratio, in Btu's per watt-hour, determined from Section 5.1 of Appendix M to this subpart; (B) the representative average use cycle of 1000 cooling load hours per year; (C) a conversion factor of .001 kilowatts per watt; and (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to Section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year;

(ii) For heat pumps which provide only heating or the heating portion of the estimated annual operating cost for central air conditioners which provide both heating and cooling, the product of (A) the quotient of the standardized design heating requirement nearest the capacity measured in the High Temperature Test, Q_{ss} (47) in Btu's per hour, determined in Sections 5.2 and 6.2.6 of Appendix M to this subpart, and the heating seasonal performance factor, in Btu's per watt-hour, calculated for heating region IV corresponding to the above mentioned standardized design heating requirement determined from Section 5.2 of Appendix M to this subpart; (B) the representative average use cycle of 2080 heating load hours per year; (C) 0.77, a correction factor described in section 5.2 of Appendix M to this subpart; (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to Section 323(b)(2) of the Act; and (E) a conversion factor of .001 kilowatts per watt; the resulting quotient then being rounded off to the nearest dollar per year; or

(iii) For central air conditioners which provide both heating and cooling, the quantity determined in paragraph (m)(1)(i) of this section added to the quantity determined in paragraph (m)(1)(ii) of this section.

(2) [Reserved]

(3) The estimated regional annual operating costs for cooling only units and air-source heat pumps shall be one of the following:

(i) For cooling only units or the cooling portion of the estimated annual operating cost for central air conditioners which provide both heating and cooling, the product of (A) the quotient of cooling capacity in Btu's per hour determined from the steady-state wet-coil test (test A) measured at the highest compressor speed, and the average seasonal energy efficiency ratio in Btu's per watt-hour both determined from Section 5.1 of Appendix M to this subpart; (B) the estimated number of local cooling load hours per year determined from Section 6.1.3 of Appendix M to this subpart; (C) a conversion factor of .001 kilowatts per watt; and (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to Section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(ii) For heat pumps which provide only heating or the heating portion of the estimated annual operating cost for central air conditioners which provide both heating and cooling, for each applicable standardized design heating requirement, the product of (A) the quotient of the estimated number of local heating load hours per year determined from Section 6.2.5 of Appendix M to this subpart, and the heating seasonal performance factor, in Btu's per watt-hour, corresponding to the appropriate standardized design heating requirement determined from Section 5.2 of Appendix M to this subpart; (B) 0.77, a correction factor described in Section 5.2 of Appendix M to this subpart; (C) the standardized design heating requirement in Btu's per hour determined in Sections 5.2 and 6.2.6 of Appendix M to this subpart; (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to Section 323(b)(2) of the Act; and (E) a conversion factor of .001 kilowatts per watt, the resulting quotient then being rounded off to the nearest dollar per year; or

(iii) For central air conditioners which provide both heating and cooling, the quantity determined in paragraph (m)(3)(i) of this section added to each quantity determined in paragraph (m)(3)(ii) of this section.

(4) [Reserved]

(5) The measure(s) of efficiency for cooling only units and air-source heat pumps shall be one or more of the following:

(i) The seasonal energy efficiency ratio for cooling only units and air-source heat pumps which provide cooling shall be the seasonal energy efficiency ratio, in Btu's per watt-hour, determined according to Section 5.1 of Appendix M to this Subpart, rounded off to the nearest 0.05.

(ii) The heating seasonal performance factors for air-source heat pumps shall be the heating seasonal performance factors, in Btu's per watt-hour, determined according to Section 5.2 of Appendix M to this subpart for each applicable standardized design heating requirement within each climatic region, rounded off to the nearest 0.05.

(iii) The annual performance factors for air-source heat pumps which provide heating and cooling, shall be the annual performance factors, in Btu's per watt-hour, determined according to Section 5.3 of Appendix M to this subpart for each standardized design heating requirement within each climatic region, rounded off to the nearest 0.05.

(6) [Reserved]

(7) Other useful measures of energy consumption for central air conditioners shall be those measures of energy consumption which the Secretary of Energy determines are likely to assist consumers in making purchasing decisions and which are derived from the application of Appendix M to this subpart.

* * * * *

3. Section 430.23 is amended by establishing a new paragraph (m) to read as follows:

§ 430.23 Units to be tested.

* * * * *

(m)(1) For central air conditioners, each condensing unit shall have a

condenser-evaporator coil combination selected and a sample of sufficient size tested in accordance with applicable provisions of this subpart such that—

(i) Any represented value of estimated annual operating cost, energy consumption or other measure of energy consumption of the condenser-evaporator coil combination for which consumers would favor lower values shall be no less than the higher of (A) the mean of the sample or (B) the upper 90 percent confidence limit of the true mean divided by 1.05, and

(ii) Any represented value of the energy efficiency or other measure of energy consumption of the condenser-evaporator coil combination for which consumers would favor higher values shall be no greater than the lower of (A) the mean of the sample or (B) the lower 90 percent confidence limit of the true mean divided by 0.95.

(2) The condenser-evaporator coil combination [1] selected for tests pursuant to paragraph (m)(1) shall be that combination manufactured by the condensing unit manufacturer likely to have the largest volume of retail sales.

For every other condenser-evaporator coil combination manufactured by the same manufacturer or in part by a component manufacturer using that same condensing unit, either—

(i) A sample of sufficient size shall be tested to insure that the requirements of paragraphs (m)(1)(i) and (m)(1)(ii) are met for such other condenser-evaporator coil combinations, or

(ii) Represented values of the measures of energy consumption shall be determined by the use of computer simulation or engineering analysis to determine the energy consuming characteristics of such basic model, as described in paragraph (m)(5) of this section.

(3) Whenever measures of energy consumption determined by computer simulation or by engineering analysis do not agree with measures of energy consumption determined by actual testing, the values determined by actual testing shall be used to comply with section 323(c) of the Act, or to comply with rules prescribed under section 324 of the Act.

(4) The sample selected for testing under paragraph (m)(1) of this section shall be comprised of units which are production units, or representative of production units.

[1] Components of similar design may be substituted without requiring additional testing if the represented measures of energy consumption continue to satisfy the applicable sampling provision.

(5) The basis of the computer simulation or engineering analysis method referred to in paragraph (m)(2)(ii) of this section shall be a representation of the mechanical vapor compression refrigeration cycle. The major components in the refrigeration cycle shall be modeled as "fits" to manufacturer performance data or by the use of graphic or tabular performance data. Heat transfer characteristics of coils may be modeled as a function of face area, number of rows, fins per inch, refrigerant circuiting, air flow rate and entering air enthalpy. An iterative process shall be used which varies system temperatures and flow rates until the cycle is balanced.

(6) When making representations regarding energy consumption, manufacturers must distinguish which condenser-evaporator coil combinations have been tested to fulfill the sampling requirements for labeling and representations and which units have been rated by computer simulation or engineering analysis. If after a period of time (not exceeding one year from introduction of a new product into the market) a manufacturer discovers that the condenser-evaporator coil combination tested does not have the largest volume of retail sales he must test the combination which, in fact, does have the largest sales volume and provide that information to the Federal Trade Commission (FTC).

(7) Manufacturers who choose to use computer simulation or engineering analysis for determing measures of energy consumption under paragraphs (m)(2)(ii) and (m)(5) of this section must—

(i) submit to DOE a written document which describes the method used to derive such ratings, and describes the procedure used by the manufacturer to verify the accuracy of such ratings; and

(ii) for every family of basic models using the same condensing unit, submit to DOE the results from all tests required by section 3 of Appendix M to this subpart used to establish the rated value(s) for the condenser-evaporator coil combination likely to have the highest sales volume, and a comparison of those test results with the values predicted when the simulation method used to rate other members of the family is used to rate such combination.

(8) Manufacturers who choose to use computer simulation or engineering analysis for determining measures of energy consumption under paragraphs (m)(2)(ii) and (m)(5) of this section shall permit representatives of the Department of Energy to inspect for verification purposes the simulation method or methods used. This inspection may include conducting simulations to predict the performance of particular condenser-evaporator coil combinations specified by DOE, analysis of previous simulations conducted by a manufacturer, or both.

§ 430.24 [Revoked]

4. Section 430.24(m) is revoked.

5. Appendix M to Subpart B of Part 430 is amended to read as follows:

Appendix M to Subpart B—Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners

1. Definitions

1.1 "Annual performance factor" means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric power used in one year.

1.2 "ARI" means Air-Conditioning and Refrigeration Institute.

1.3 "ARI Standard 210–79" means the test standard published in 1979 by the ARI and titled "Standard for Unitary Air-Conditioning Equipment".

1.4 "ARI Standard 240–77" means the test standard published in 1977 by the ARI and titled "Standard for Air-Source Unitary Heat Pump Equipment".

1.5 "ARI Standard 320–76" means the test standard published in 1976 by the ARI and titled "Standard for Water-Source Heat Pumps".

1.6 "ASHRAE" means the American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

1.7 "ASHRAE Standard 37–78" means the test standard published by ASHRAE in 1978 and titled "Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment."

1.8 "Continuously recorded" means a method of recording measurements in intervals no greater than 5 seconds.

1.9 "Cooling load factor (CLF)" means the ratio of the total cooling done in a complete cycle of a specified time period, consisting of an "on" time and "off" time, to the steady-state cooling done over the same period at constant ambient conditions.

1.10 "Cyclic Test" means a test where the indoor and outdoor conditions are held constant, but the unit is manually turned "on" and "off" for specific time periods to simulate part-load operation.

1.11 "Degradation coefficient (C_D)" means the measure of the efficiency loss due to the cycling of the unit.

1.12 "Demand-defrost control system" means a system which is designed to perform the defrost function on the outdoor coil of the heat pump only when a predetermined degradation of performance is measured.

1.13 "Design heating requirement (DHR)" is the amount of heating required to maintain a given indoor temperature at a particular outdoor design temperature.

1.14 "Dry-coil test" means a test conducted at a wet-bulb temperature and a dry-bulb temperature such that moisture will not condense on the evaporator coil of the unit.

1.15 "Heating seasonal performance factor (HSPF)" means the total heating output of a heat pump during its normal annual usage period for heating divided by the total electric power input during the same period.

1.16 "Heating load factor (HLF)" means the ratio of the total heating done in a complete cycle of a specified time period, consisting of an "on" time "off" time, to the steady state heating done over the same period at constant ambient conditions.

1.17 "Latent cooling" means the amount of cooling in Btu's necessary to remove water vapor from the air passing over the indoor coil by condensation during a period of time.

1.18 "Part-load factor (PLF)" means the ratio of the cyclic energy efficiency ratio to the steady-state energy efficiency ratio at identical ambient conditions.

1.19 "Seasonal energy efficiency ratio (SEER)" means the total cooling of a central air conditioner in Btu's during its normal annual usage period for cooling divided by the total electric power input in watt-hours during the same period.

1.20 "Sensible cooling" means the amount of cooling in Btu's performed by a unit over a period of time, excluding latent cooling.

1.21 "Single package unit" means any central air conditioner in which all the major assemblies are enclosed in one cabinet.

1.22 "Split system" means any central air conditioner in which one or more of the major assemblies are separate from the others.

1.23 "Steady-state test" means a test in which all indoor and outdoor conditions are held constant and the unit is in non-changing operating mode.

1.24 "Temperature bin" means a 5° F increment over a dry-bulb temperature range of 65° F through 104° F for the cooling cycle and −25° F through 64° F for the heating cycle.

1.25 "Time-temperature defrost control system" means a system which automatically provides the defrost function at a predetermined time interval whenever the outdoor temperature drops below a level where frosting will occur.

1.26 "Test condition tolerance" means the maximum permissible variation of the average of the test observations from the standard or desired test condition as provided in 6.1.1, 6.2.1, 6.2.2, and 6.2.3 of this Appendix.

1.27 "Test operating tolerance" means the maximum permissible difference between the maximum and the minimum instrument observation during a test as provided in 6.1.1, 6.2.1, 6.2.2, and 6.2.3 of this Appendix.

1.28 "Wet-coil test" means a test conducted at a wet-bulb temperature and a dry-bulb temperature such that moisture will condense on the test unit evaporator coil.

2. Testing Required

2.1 *Testing required for air source cooling only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* Two steady state wet coil tests required to be performed, test A and test B. Test A is to be conducted as an outdoor dry bulb temperature of 95° F and test B at 82° F. Test C and D are optional tests to be conducted when cyclic performance parameters are to be measured in order to determine the degradation coefficient, C^D Test C is a steady state dry coil test conducted at an outdoor dry bulb temperature of 82° F. Test D is a cyclic test also conducted at an outdoor dry bulb temperature of 82° F. In lieu of conducting tests C and D, an assigned value of 0.25 may be used for the degradation coefficient, C^D.

2.1.1 *Testing required for units with single speed compressors and single speed condenser fans.* Test A and test B shall be performed according to the test procedures outlined in 4.1 of this Appendix. In addition, the cyclic performance shall be evaluated by conducting test C and D according to the requirements outlined in 4.1 of this Appendix.

2.1.2 *Testing required for units with single speed compressors and multiple-speed condenser fans.* The test requirements for multiple-speed condenser fan units shall be the same as described in section 2.1.1 for single speed condensor fan units.

2.1.3 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading.* The test requirements for two-speed compressor units, two compressor units, or units with cylinder unloading are the same as described in 2.1.1 of this Appendix except that test A and test B shall be performed at each compressor speed or at each compressor capacity.

2.1.4 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading capable of varying the sensible to total (S/T) capacity ratio.* When a unit employing a two-speed compressor, two compressors, or cylinder unloading provides a method of varying the ratio of the sensible cooling capacity to the total cooling capacity, (S/T), the test requirements are the same as for two-speed compressor units as described in 2.1.3 of this Appendix.

2.2 *Testing required for air source heating only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* Four types of tests are required to be performed: High Temperature, Cyclic, Frost Accumulation, and Low Temperature. In lieu of conducting the Cyclic Test an assigned value of 0.25 may be used for the degradation coefficient, C^D.

2.2.1 *Testing required for units with single speed compressors.* Units with single speed compressors shall be subjected respectively to the High Temperature Test at 47° F described in section 3.2.1.1, the Cyclic Test as described in section 3.2.1.2, the Frost Accumulation Test as described in section 3.2.1.3, and the Low Temperature Test as described in section 3.2.1.4.

2.2.2 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading.* With the unit operating: at high compressors speed (two-speed compressor), with both compressors in operation (two-compressors), or at the maximum capacity (cylinder unloading); the following tests are required to be performed on all units; the High Temperature Test at 47° F, the Frost Accumulation Test, and the Low Temperature Test. An additional test (cyclic at 47° F) is required, with the unit operating at the high compressor speed (two-speed compressor), with both compressors in operation (two compressors), or at the maximum capacity (cylinder unloading); if the normal mode of operation requires cycling "on" and "off" of the compressor(s) at high speed or maximum capacity.

With the unit operating: at the low compressor speed (two-speed compressor), with the single compressor which normally operates at low loads (two compressors), or at the low compressor capacity (cylinder unloading); the following tests are required to be performed on all units: the High Temperature Test at 47° F, the High Temperature Test at 62° F, and the Cyclic Test. Additional tests, (Frost Accumulation Test and Low Temperature Test) are required, with the unit operating: on low compressor speed (two-speed compressor), with the single compressor which normally operates at low loads (two compressors) or at the low compressor capacity (cylinder unloading), if the unit's low speed, one compressor or low capacity performance at and below 40° F is needed to calculate its seasonal performance.

2.3 *Testing required for air source units which provide both heating and cooling.* The requirements for units which provide both heating and cooling shall be the same as the requirements in Section 2.1. and 2.2 of this Appendix.

2.4 *Testing required for water source units which provide both heating and cooling.* (Reserved)

3. Testing conditions

3.1 *Testing conditions for air source cooling only units with single speed compressors, two-speed compressors, two compressors or cylinder unloading.* The test room requirement and equipment installation procedures are the same as those specified in sections 11.1 and 11.2 of ASHRAE Standard 37–78. Units designed for both horizontal and vertical installation shall be tested in the orientation in which they are most frequently installed. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 or 230 volts and 60 hertz), the test installation shall be designed such that there will be no air flow through the cooling coil due to natural or forced convection while the indoor fan is "off". This shall be accomplished by installing dampers upstream and downstream of the test unit to block the off period air flow. Values of capacity for rating purposes are to be rounded off to the nearest 100 Btu/hour for capacities less than 20,000 Btu/hour; to the nearest 200 Btu/hour for capacities between 20,000 and 37,999 Btu/hour; and to the nearest 500 Btu/hour for capacities between 38,000 and 64,999 Btu/hour.

The following conditions listed in ARI Standard 210.79 shall apply to all tests performed in Section 3.1 of this Appendix:

5.1.3.4 Cooling Coil Air Quantity.

5.1.3.6 Requirements for Separated Assemblies.

3.1.1 *Testing conditions for units with single speed compressors and single speed condenser fans.*

3.1.1.1 *Steady state wet-coil performance tests (Test A and Test B).* Test A and test B shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80° F and a wet-bulb temperature of 67° F. The dry-bulb

temperature of the air entering the outdoor side of the unit shall be 95° F in test A and 82° F in test B. The temperature of the air surrounding the outdoor side of the unit in each test shall be the same as the outdoor entering air temperature except for units or sections thereof intended to be installed only indoors, in which case the dry-bulb temperature surrounding that indoor side of the unit shall be 80° F. For those units which reject condensate to the condenser, located in the outdoor side of the unit, the outdoor wet-bulb temperature surrounding the outdoor side of the unit shall be 75° F in test A and 65° F in test B.

3.1.1.2 *Steady state dry coil performance test (Test C) and cyclic dry coil performance test (Test D).* Test C and test D shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80° F and a wet-bulb temperature which does not result in formation of condensate on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57° F or less be used.) The dry-bulb temperature of the air entering the outdoor portion of the unit shall be 82° F. The outdoor portion of the unit shall be subject to the same conditions as the requirements for conducting test B as stated previously in section 3.1.1.1. Test C shall be conducted with the unit operating steadily. Test D shall be conducted by cycling the unit "on" and "off" by manual or automatic operation of the normal control circuit of the unit. The unit shall cycle with the compressor "on" for 6 minutes and "off" for 24 minutes. The indoor fan shall also cycle "on" and "off", the duration of the indoor fan "on" and "off" periods being governed by the automatic controls which the manufacturer normally supplies with the unit. The results of tests C and D shall be used to calculate a degradation coefficient, C_D by the procedures outlined in 5.1 of this Appendix.

3.1.2 *Testing conditions for units with single speed compressors and multiple-speed condenser fans.* The condenser fan speed to be used in test A shall be that speed which normally occurs at an outdoor dry-bulb temperature of 95° F, and for test B, the fan speed shall be that which normally occurs at an outdoor dry-bulb temperature of 82° F. If elected to be performed, tests C and D shall be conducted at the same condenser fan speed as in test B.

3.1.3 *Testing conditions for units with two-speed compressors, two compressors, or cylinder unloading.* The condenser fan speed used in conducting test A at each compressor speed shall be that which normally occurs at an outdoor dry-bulb temperature of 95° F. For test B, the condenser fan speed at each compressor speed shall be that which normally occurs at an outdoor dry-bulb temperature of 82° F. If elected to be performed, tests C and D shall be conducted at the low compressor speed with the same condenser fan speed as used in test B. For those two-speed units in which the normal mode of operation involves cycling the compressor "on" and "off" at high speed, tests C and D shall also be performed with the compressor operating at high speed and at a condenser fan speed that normally occurs at test A ambient conditions. Units consisting of two compressors are subject to the same requirements as those units containing two-speed compressors, except that when operated at high speed, both compressors shall be operating and when operating at low speed, only the compressor which normally operates at an outdoor dry-bulb temperature of 82° F shall be operating.

In lieu of conducting tests C and D, an assigned value of 0.25 may be used for the degradation coefficient, C_D, at each compressor speed. If the assigned degradation coefficient is used for one compressor speed it must also be used for the other compressor speed.

In the case of units with cylinder unloading, the loaded and the unloaded conditions correspond to high and low compressor speed on two-speed units respectively.

3.1.4 *Testing conditions for units with two-speed compressors, two compressors, or cylinder unloading capable of varying the sensible to total (S/T) capacity ratio.* The mode of operation selected for controlling the S/T ratio in the performance of test A and test B at each compressor speed shall be such that it does not result in an operating configuration which is not typical of a normal residential installation. If elected to be performed, tests C and D shall be conducted at low compressor speed (single compressor operating) with the same S/T control mode as used in test B when performed at the low compressor speed. Likewise, tests C and D shall also be conducted at high compressor speed (two compressors operating) and with the same S/T control mode as in test A when performed at the high compressor speed.

In the case of units with cylinder unloading, the loaded and unloaded conditions correspond to high and low compressor speed on two-speed units respectively.

3.2 *Testing conditions for air source heating only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* The equipment under test shall be installed according to the requirements of Section 11.2 of ASHRAE Standard 37–78 and Section 5.1.4.5 of ARI Standard 240–77. Test chamber requirements are the same as given in Section 11.1 of ASHRAE Standard 37–78. Units designed for both horizontal and vertical installation shall be tested in the orientation in which they are most often installed. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 or 230 volts and 60 hertz). Values of capacity for rating purposes are to be rounded off to the nearest 100 Btu/hour for capacities less than 20,000 Btu/hour; to the nearest 200 Btu/hour for capacities between 20,000 and 37,999 Btu/hour; and to the nearest 500 Btu/hour for capacities between 38,000 and 64,999 Btu/hour.

3.2.1 *Testing conditions for units with single speed compressors.*

3.2.1.1 *High temperature test conditions.* The High Temperature Test at 47° F shall be conducted at an outdoor dry-bulb temperature of 47° F and an outdoor wet-bulb temperature at 43° F. The High Temperature Test at 62° F shall be conducted at an outdoor dry-bulb temperature of 62° F and an outdoor wet-bulb temperature of 56.5° F. For both tests, the dry-bulb air temperature entering and surrounding the indoor portion of the unit shall be 70° F and a maximum wet-bulb temperature of 60° F. The duration of the tests shall be for a minimum of ½ hour.

3.2.1.2 *Cycling test conditions.* The Cycling Test at 47° F shall be conducted at the same dry-bulb and wet-bulb temperature as the High Temperature Test at 47° F as described in 3.2.1.1. During the Cycling Test, the indoor fan shall cycle "on" and "off", as the compressor cycles "on" and "off", except that the indoor fan cycling times may be delayed due to controls that are normally installed with the unit. The compressor cycling times shall be 6 minutes "On" and 24 minutes "off." The test installation shall be designed such that there will be no airflow through the indoor unit due to natural or forced convection while the indoor fan is "off." This shall be accomplished by installing dampers upstream and downstream of the test unit to block the off period airflow.

3.2.1.3 *Frost accumulation test conditions.* The dry-bulb temperature and the resultant dew-point temperature of the air entering the outdoor portion of the unit shall be 35° F and 30° F respectively. The indoor dry-bulb temperature shall be 70° F and the maximum indoor wet-bulb temperature shall be 60° F. The Frost Accumulation Test requires that the unit undergo a defrost prior to the actual test. The test then begins at defrost termination and ends at the next defrost termination. Defrost termination occurs when the controls normally installed within the unit are actuated to cause it to change defrost operation to normal heating operation. During the test, auxiliary resistance heaters shall not be employed during either the heating or defrost portion of the test.

3.2.1.4 *Low temperature test conditions.* The Low Temperature Test shall be conducted at a dry-bulb temperature entering the outdoor portion of the unit of 17° F and a wet-bulb temperature of 15° F. The air entering the indoor portion of the unit shall have a dry-bulb temperature of 70° F and a maximum wet-bulb temperature of 60° F.

3.2.1.5 *Additional testing conditions.* All tests shall be conducted at the indoor-side air quantities specified in Sections 4.1.4.3 and 5.1.4.6 and Table 2 of ARI Standard 240–77. The following conditions listed in ARI Standard 240–77 shall apply to all tests performed in Section 3.2 of this Appendix.

5.4.4.4 *Outdoor-Side Air Quantity; and*
5.1.4.5 *Requirements for Separated Assemblies.*

In all tests, the specified dry-bulb temperature entering the outdoor portion of the unit also applies to the air temperature surrounding the outdoor portion of the unit. Similarly, models where portions are intended to be installed indoors shall have the air temperature surrounding that portion of the unit the same as the indoor air temperature.

3.2.2 *Testing conditions for units with two-speed compressors, two compressors or cylinder unloading.* The testing conditions for two-speed compressors, two compressors, or cylinder unloading shall be the same as those for single speed units as described in 3.2.1.

3.3 *Testing conditions for air source units which provide both heating and cooling.* The testing conditions for units which provide both heating and cooling shall be the same as the requirements in Sections 3.1 and 3.2 of this Appendix.

3.4 *Testing conditions for water source units which provide both heating and cooling.* (Reserved)

4.0 *Testing procedures.* Measure all electrical inputs as described in the procedures below. All electrical measurements during all "on" and "off" periods shall include auxiliary power or energy (controls, transformers, crankcase heaters, etc.) delivered to the unit.

4.1 *Testing procedures for air source cooling only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* All steady-state wet- and dry-coil performance tests on single package units shall simultaneously employ the Air-Enthalpy Method (Section 3 of ASHRAE Standard 37–78) on the indoor side and one other method consisting of either the Air-Enthalpy Method or the Compressor Calibration Method (Section 4 of ASHRAE Standard 37–78 on the outdoor side. All steady-state wet- and dry-coil performance tests on split systems shall simultaneously employ the Air-Enthalpy Method or the Compressor Calibration Method on the indoor side and the Air-Enthalpy Method, the Compressor Calibration Method or the Volatile Refrigerant Flow Method (Section 5 of ASHRAE Standard 37–78) on the outside. All cyclic dry-coil performance tests shall employ the Air-Enthalpy Method, indoor side only. The values calculated from the two test methods must agree within 6 percent in order to constitute a valid test. Only the results from the Air-Enthalpy Method on the indoor side shall be used in the calculations in Section 5.1. Units shall be installed and tested in such a manner that when operated under steady-state conditions, the cooling coil and condenser coil air flows meet the requirements of Sections 5.1.3.4, 5.1.3.5, and 5.1.3.7 of ARI Standard 210.79.

4.1.1 *Test operating procedures.*

4.1.1.1 *Steady-state wet-coil performance tests (Test A and Test B).* Steady-state wet-coil performance tests (A and B) shall be conducted in accordance with the conditions described in sections 3.1.1.1, 3.1.2, 3.1.3, 3.1.4, and 3.1.5 of this Appendix and the procedures described for cooling tests in Section 11.3 of ASHRAE standard 37–78 and evaluated in accordance with the cooling-related requirements of Section 12 of the ASHRAE Standard 37–78. The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained.

4.1.1.2 *Steady-state and cyclic dry-coil performance tests (Test C and Test D).* The steady-state and cyclic dry-coil tests (C and D) shall be conducted as described below in accordance with the conditions described in sections 3.1.1.2, 3.1.2, 3.1.3, 3.1.4, and 3.1.5 of this Appendix. The results shall be evaluated in accordance with the cooling related requirements of Sections 12.1.5, 12.1.6, 12.1.7, of ASHRAE Standard 37.78. The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but not for less than one hour before data for test C are recorded. For all equipment test methods including the Compressor Calibration Method, test C shall be performed with data recorded at 10-minute intervals until four consecutive sets of readings are attained with the tolerance prescribed in Section 11.6 of ASHRAE Standard 37–78. When the Air-Enthalpy Method is used on the outdoor side for test C, the requirements of this section shall apply to both the preliminary test and the regular equipment test; the requirements of Section 3.6 of ASHRAE Standard 37–78 shall also apply. Immediately after test C is completed the test unit shall be manually cycled "off" and "on" using the time periods from 3.1.1 of this Appendix until steadily repeating ambient conditions are again achieved in both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be run through an additional "off"/"on" cycle during which the test data required in 5.1 shall be recorded. During this last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerances specified in 4.1.3 of this Appendix during the cyclic dry-coil tests, all air moving equipment on the condenser side shall cycle "on" and "off" when the compressor cycles "on" and "off". The indoor air moving equipment shall also cycle "off" as governed by any automatic controls normally installed with the unit. This last requirement applies to units having an indoor fan time delay. Units not supplied with an indoor fan time delay shall have the indoor air moving equipment cycle "on" and "off" as the compressor cycles "on" and "off."

4.1.2 *Test instrumentation.* The steady-state and cyclic performance tests shall have the same requirements pertaining to instrumentation and data as those specified in Section 10 and Table II of ASHRAE Standard 37.78. For the cyclic dry-coil performance tests, the dry-bulb temperature of the air entering and leaving the cooling coil, or the difference between these two dry-bulb temperatures, shall be continuously recorded with instrumentation accurate to within ±0.3° F of indicated value and have a response time of 2.5 seconds or less. Response time in the time required for the instrumentation to obtain 63 percent of the final steady-state temperature difference when subjected to a step change in temperature difference of 15° F or more. Electrical measurement devices (watt-hour meters) used during all tests shall be accurate to within ±0.5 percent of indicated value.

4.1.3 *Test tolerances.* All steady-state wet-and dry-coil performance tests shall be performed within the applicable operating and test condition tolerances specified in Section 11.6 and Table III of ASHRAE Standard 37–78.

4.1.3.1 The indoor and outdoor average dry-bulb temperature for the cyclic dry coil test D shall both be within 1.0° F of the indoor and outdoor average dry bulb temperature for the steady-state dry coil test C, respectively.

4.1.3.2 The test condition and test operating tolerances for conducting test D are stated in 6.1.1 of this Appendix. Variation in the test conditions greater than the tolerances prescribed in 6.1.1 of this Appendix shall invalidate the test. It is suggested that an electric resistance heater having a heating capacity approximately equal to the sum of the cooling capacity and compressor and condenser fan power should be installed in the outdoor test room and cycled "off" and "on" as the unit cycles "on" and "off" respectively to improve control in the outdoor test room. Similarly, an electric resistance heater having a heating capacity approximately equal to the cooling capacity of the unit could be installed in the indoor test room, and cycled "on" and "off" as the test unit cycles "on" and "off" to improve indoor room control.

4.2 *Testing procedures for air source heating only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.*

4.2.1 *Test operating procedures.* All High Temperature Tests, the Cyclic Test, the Frost Accumulation Test, and the low Temperature test shall have the performance evaluated by the Air-Enthalpy Method on the indoor side. In addition, the High Temperature Test and the Low Temperature Test shall have a simultaneous test method (as described in 4.1) used as a check. The values calculated from the two methods must agree within 6 percent in order to constitute a valid test. Only the results from the Air-Enthalpy Method on the indoor side shall be used in the calulations in section 5.2.

4.2.1.1 *Test procedure for high temperature test.* When the outdoor Air-Enthalpy Method is used, the outdoor chamber must not interfere with the normal air circulating pattern during the preliminary test. It is necessary to determine and adjust for system resistance when the outdoor air measuring apparatus is attached to the outdoor portion of the unit. The test room apparatus and test units must be operated for at least one hour with at least ½ hour at equilibrium and at the specified test conditions prior to starting the test. The High Temperature Test shall then be conducted for a minimum of ½ hour with intermittent data being recorded at 10-minute intervals. For all units, especially those having controls which periodically cause the unit to operate in defrost mode, attention should be given to prevent defrost during the High Temperature Test. Units which have undergone a defrost should operate in the heating mode for at least 10-minutes after defrost termination prior to the start of the test. When the outdoor Air-Enthalpy Method is used as a second test then a preliminary test must be conducted for a minimum of 30 minutes with 4 or more sets of data recorded at 10 minute intervals, all remaining requirements of Section 3.6.1 in the ASHRAE Standard 37–78 shall then apply in conducting the preliminary test for the outdoor air enthalpy method. For some units, at the ambient condition of the test, frost may accumulate on the outdoor coil. If the supply air temperature or the difference between the supply air temperature and the indoor air entering temperature has decreased by more than 1.5° F at the end of the test, the unit shall be defrosted and the test restarted. Only the

results of this second High Temperature Test shall be used in the heating seasonal performance calculation in section 5.2. Prior to beginning the High Temperature Test, a unit shall operate in the heating mode for at least 10 minutes after defrost termination to establish equilibrium conditions for the unit and the room reconditioning apparatus. The High Temperature Test may only begin when the test unit and room conditions are within the test condition tolerances specified in Section 6.2.1 of this Appendix.

4.2.1.2 *Test procedures for the cyclic test.* The cyclic test shall follow the High Temperature Test and by cycled "on" and "off" as specified in 3.2.1.2 until steadily repeating ambient conditions are achieved for both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be operated through an additional "off"/"on" cycle, during which the required test data shall be recorded. During the last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerance specified in section 6.2.2. of this Appendix. If, prior to the High Temperature Test, the unit underwent a defrost cycle to rid the outdoor coil of any accumulated frost, then prior to cycling the unit "off" and "on" it should be made to undergo a defrost. After defrost is completed and before starting the cycling process, the unit shall be operated continuously in the heating mode for a least 10 minutes to assure that equilibrium conditions have again been established for the unit and the room conditioning apparatus. Cycling the unit may begin when the test unit and room conditions are within the High Temperature Test condition tolerances specified in section 6.2.1 of this Appendix. Attention should be given to prevent defrost after the cycling process has begun.

4.2.1.3 *Test procedures for the frost accumulation test.* The defrost controls shall be set at the normal settings which most typify those encountered in Region IV as described in section 6.2.4 and 6.2.5 of this Appendix. The test room reconditioning equipment and the unit under test shall be operated for at least ½ hour prior to the start of a "preliminary" test period. The preliminary test period and the test period itself are to be conducted within the test tolerances given in section 4.2.3.3 of this Appendix. In some cases, the preliminary defrost cycle may be manually induced, however, it is important that the normally operating controls govern the defrost termination in all cases. For units containing defrost controls which are likely to cause defrost at intervals less than one hour when the unit is operating at the required test conditions, the preliminary test period shall start at the termination of a defrost cycle which automatically occurs and shall end at the termination of the next automatically occurring defrost cycle. For units containing defrost controls which are likely to cause defrost at intervals exceeding one hour when operating at the required test condition, the preliminary test period consists of "heating-only" preliminary operation for at least one hour, after which a defrost may be manually or automatically induced. The test period then begins at the termination of this defrost cycle and ends at the termination of the next automatically occurring defrost cycle. If the unit has not undergone a defrost after 12 hours, then the tests shall be concluded and the results calculated for this 12-hour period. For units which turn the indoor fan off during defrost the indoor supply duct shall be blocked during all defrost cycles to prevent natural or forced convection through the indoor unit. During defrost, resistance heaters normally installed with the unit shall be prevented from operating.

4.2.1.4 *Test procedures for the low temperature test.* Where applicable, the High Temperature Test preparation and performance requirements shall also be used in the Low Temperature Test. The test room reconditioning equipment shall first be operated in a steady-state manner for at least one-half hour at equilibrium and at the specified test conditions. The unit shall then undergo a defrost, either automatic or manually induced. It is important that the unit terminate the defrost sequence by the action of its own defrost controls. The defrost controls are to remain at the same setting as specified in 4.2.1.3. At a time no earlier than 10 minutes after defrost termination, the test shall start. Test duration is one-half hour. For all units, defrost should be prevented during the one-half hour test period.

4.2.2 *Test instrumentation.*

4.2.2.1 *Test instrumentation for the high temperature test.* The indoor air flow rate shall be determined as described in Section 7.1 through 7.4 of ASHRAE Standard 37–78. This requires the construction of an air receiving chamber and discharge chamber separated by partition in which one or more nozzles are located. The receiving chamber is connected to the indoor air discharge side of the test specimen through a short plenum. The exhaust side of the air flow rate measuring device contains an exhaust fan with some means to vary its capacity to obtain the desired external resistance to air flow rate. The exhaust side is then left open to the test room or is ducted through a conditioning apparatus and then back to the test specimen inlet. The static pressure across the nozzles, the velocity pressure, and the static pressure measurements at the nozzle throat shall be measured with manometers which will result in errors which are no greater than ±1.0 percent of indicated value and having minimum scale divisions not exceeding 2.0 percent of the reading. Static pressure and temperature measurements must be taken at the nozzle throat in order to obtain density of the air. The areas of the nozzles shall be determined by measuring their diameter with an error no greater than ±0.2 percent in four places approximately 45 degrees apart around the nozzle in each of two places through the nozzle throat, one at the outlets and the others in the straight section near the radius. The energy usage of the compressor, indoor and outdoor fan, and all other equipment components shall be measured with a watt-hour meter which is accurate to within ±0.5 percent of the quantity measured. Measurements of the air temperature entering and leaving the indoor coil or the difference between these two shall be made in accordance with the requirements of ASHRAE Standard 41 part 1. These temperatures shall be continuously recorded with instrumentation having a total system accuracy within ±0.3° F of indicated value and a response time of 2.5 seconds or less. Temperature measurements are to be made upstream of the static pressure tap on the inlet and downstream of the static pressure taps on the outlet. The indoor and outdoor dry-bulb temperatures shall be continuously recorded with instrumentation which will result in an error no greater than ±0.3° F of indicated value. The outdoor wet-bulb temperature shall be continuously recorded. Static pressure measurements in the ducts and across the unit shall be made in accordance with Section 8 of ASHRAE Standard 37–78 using equipment which will result in an error no greater than ±0.01 inch of water. Static pressure measurements shall be made and recorded at 5 minute intervals. All other data not continuously recorded shall be recorded at 10 minute intervals.

4.2.2.2 *Test instrumentation for the cycling test.* The air flow rate during the on-period of the Cyclic Test shall be the same agreed within ±1. percent as the air flow rate measured during the previously conducted High Temperature Test. All other instrumentation requirements are identical to 4.2.2.1 of this Appendix.

4.2.2.3 *Test instrumentation for the frost accumulation test.* The air flow rate for the Frost Accumulation Test shall be the same as described in 4.2.2.1. The indoor-side dry-bulb temperature and outdoor-side dry-bulb temperature shall be continuously recorded with instrumentation having a total system accuracy within ±0.3° F of indicated value. The outdoor dew point temperature shall be determined with an error no greater than ±0.5° F of indicated value using continuously recording instrumentation. All other data shall be recorded at 10 minute intervals during the heating cycle. Defrost initiation, termination and complete test cycle time (from defrost termination to defrost termination) shall be recorded. Defrost initiation is defined as the actuation (either automatically or manually) of the controls normally installed with the unit which cause it to alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil. Defrost termination occurs when the controls normally within the unit are actuated to change from defrost operation to normal heating operation. Provisions should be made so that instrumentation in capable of recording the cooling done during defrost as well as the total electrical energy usage during defrost. These data and the continuously recorded data need be the only data obtained during defrost.

4.2.2.4 *Test instrumentation for the low temperature test.* Instrumentation for the Low Temperature Test is identical to that of the High Temperature Test described in section 4.2.2.1 of this Appendix.

4.2.3 *Test tolerances.*

4.2.3.1 *Test tolerances for the high temperature test.* All tests shall be conducted within the tolerances specified in Section 6.2.1. Variation greater than those given shall

invalidate the test. The heating capacity results by the indoor Air Enthalpy Method shall agree within 6 percent of the value determined by any other simultaneously conducted capacity test in order for the test to be valid.

4.2.3.2 *Test tolerances for the cyclic test.* The test condition tolerances and test operating tolerances for the on-period portion of the test cycle are specified in Section 6.2.2. Variation exceeding any specified test tolerance shall invalidate the test results.

4.2.3.3 *Test tolerances for the frost accumulation test.* Test condition and test operating tolerances for Frost Accumulation Tests are specified in Section 6.2.3. Test operating tolerances during heating applies when the unit is in the heating mode, except for the first 5 minutes after the termination of a defrost cycle. Test operating tolerance during defrost applies during a defrost cycle and during the first 5 minutes after defrost termination when the unit is operating in the heating mode. In determining whether the test condition tolerances are met, only the heating portion of the test period shall be used in calculating the average values. Variations exceeding the tolerances presented in Section 6.2.3 shall invalidate the test.

4.2.3.4 *Test tolerances for the low temperature test.* During the test period for the Low Temperature Test, the operating conditions shall be within the tolerances specified in Section 6.2.1 of this Appendix.

4.3 *Testing procedures for air source units which provide both heating and cooling.* The testing procedures for units which provide both heating and cooling shall be the same as those specified in Sections 4.1 and 4.2 of this Appendix. Also during the off-period of the dry-coil cooling test (test D), the switch-over valve shall remain in the cooling mode, unless the controls normally supplied with the unit are designed to reverse it, in which case the controls shall operate the valve. During the off-period of the cyclic heating test at 47° F, the switch-over valve shall remain in the heating mode, unless the controls normally supplied with the unit are designed to reverse it, in which case the controls shall operate the valve.

4.4 *Testing procedures for water source units which provide both heating and cooling.* (Reserved)

5.0 *Calculations for performance factors.*

5.1 *Calculations of seasonal energy efficiency ratios (SEER) in air-source units.*

The testing data and results required to calculate the seasonal energy efficiency ratio (SEER) in Btu's per watt-hour shall include the following:

(i) Cooling capacities (Btu/hr) from tests A and B and, if applicable, the cooling capacity (Btu/hr) from test C and the total cooling done from test D (Btu's).

$\dot{Q}_{ss}^{k}$ (95F)
$\dot{Q}_{ss}^{k}$ (82F)
$\dot{Q}_{ss}$, dry
Q_{cyc}, dry

(ii) Electrical power input to all components and controls (watts) from tests A, B, and if applicable the electrical power input to all components and controls (watts) from test C and the electrical usage (watt-hour) from test D.

$\dot{E}_{ss}^{k}$ (95F)
$\dot{E}_{ss}^{k}$ (82F)
$\dot{E}_{ss}$, dry
E_{cyc}, dry

(iii) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water).

(iv) Air temperature (° F)
Outdoor dry bulb
Outdoor wet bulb
Indoor dry bulb
Indoor wet bulb

Where the cooling capacities $\dot{Q}_{ss}^{k}$ (95F), from test A, $\dot{Q}_{ss}^{k}$ (82), from test B, and $\dot{Q}_{ss}$, dry, from test C, are calculated using the equations specified in section 3.7 of ASHRAE Standard 37–78. The total cooling done, Q_{cyc}, dry from test D, is calculated using equation (1) below.

Units which do not have indoor air circulating fans furnished as part of the model shall have their measured total cooling capacities adjusted by subtracting 1250 Btu/hr per 1,000 CFM of measured indoor air flow and adding to the total steady-state electrical power input 365 watts per 1,000 CFM of measured indoor air flow.

Energy efficiency ratios from tests A, B, and C, EER_A, EER_B, EER_{ss}, dry respectively, are each calculated as the ratio of the total cooling capacity in Btu/hr to the total electrical power input in watts.

Units which do not have indoor air circulating fans furnished as part of the model shall adjust their total cooling done and energy used in one complete cycle for the effect of circulating indoor air equipment power. The value to be used for the circulating indoor air equipment power shall be 1250 Btu/hr per 1,000 CFM of circulating indoor air. The energy usage required in one complete cycle required for indoor air circulation is the product of the circulating indoor air equipment power and the duration of time in one cycle that the circulating indoor air equipment is on. The total cooling done shall then be the measured cooling in one complete cycle minus the energy usage required for indoor air circulation in one complete cycle. The total electrical energy usage shall be the sum of the energy usage required for indoor air circulation in one complete cycle and the energy used by the remaining equipment components (compressor(s), outdoor fan, crankcase heater, transformer(s), etc.) in one complete test cycle.

Energy efficiency ratio from tests D, EER_{cyc} dry is calculated as the ratio of the total cooling done in Btu's to the total electrical energy usage in watt-hours.

The results of the cyclic and steady-state dry-coil performance tests shall be used in the following (4) equations:

$$(1)\qquad Q_{cyc.\,dry}=\frac{60\times\dot{V}\times C_{pa}\times\Gamma}{[V_n'\times(1+W_n)]}$$

where

$Q_{cyc.\,dry}$ = Total cooling over a cycle consisting of one compressor "off" period and one compressor "on" period (Btu's).

$\dot{V}$ = Indoor air flow rate (cfm) at the dry-bulb temperature, humidity ratio, and pressure existing in the region of measurement.

C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air, (Btu/lb-°F).

V_n' = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).

W_n = Humidity ratio (lb/lb).

and Γ (hr.-°F), which is described by the equation:

(2)

$$\Gamma=\int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}}[T_{a1}(t)-T_{a2}(t)]dt$$

where

$T_{a1}(t)$ = Dry bulb temperature of air entering the indoor coil (°F) at time (t).

$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (°F) at time (t).

$$(3)\qquad CLF=\frac{Q_{cyc.\,dry}}{Q_{ss.\,dry}\times\tau}$$

where

CLF = cooling load factor.

$Q_{ss.\,dry}$ = Total steady-state cooling capacity from test C (Btu/hr).

τ = Duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

The preceding equations are then used in the following equation to calculate a degradation coefficient C_D rounded to the nearest .01.

$$(4)\qquad C_D=\frac{1-\frac{EER_{cyc.\,dry}}{EER_{ss.\,dry}}}{1-CLF}$$

where

$EER_{ss.\,dry}$ = Energy efficiency ratio from test C, (Btu/watt-hr).

5.1.1 Method for calculating a SEER for units with single-speed compressor and single-speed condenser fans. The seasonal energy efficiency ratio for units employing single-speed compressors and single-speed condenser fans shall be based on the performance of test B and a method outlined in 2.1.1 and 3.1.1 of this Appendix to account for the cyclic performance.

The seasonal energy efficiency ratio in Btu's/watt-hour shall be determined by the equation:

$$SEER=PLF(0.5)\times EER_B$$

where

EER_B = Energy efficiency ratio determined from test B as outlined in 2.1.1.

$PLF(0.5)$ = Part-load performance factor when cooling load factor = 0.5 as determined from the equation:

$$PLF(0.5)=1-0.5\times C_D$$

where

C_D = Is the degradation coefficient described in 2.1.1 or as calculated in equation (4) above.

5.1.2 Method for calculating a SEER for units with single-speed compressors and multi-speed condenser fans. The seasonal energy efficiency ratio (SEER) for units employing single-speed compressors and multi-speed condenser fans shall be based on the energy efficiency ratio obtained for test B and the method outlined in 2.1.2 of this Appendix to account for the performance under cyclic conditions. The energy efficiency ratio for test B is obtained with the unit operating with the condenser fan speed which normally occurs at test B ambient conditions.

The seasonal energy efficiency ratio in Btu's/watt-hour shall be determined by the equation:

$$SEER=PLF(0.5)\times EER_B$$

where

EER_B = energy efficiency ratio determined from test B in 2.8

$PLF(0.5)$ = Part-load performance factor as determined from the equation:

$$PLF(0.5)=1-0.5\times C_D$$

where

C_D = The degradation coefficient described in 2.1.2 or as calculated in equation (4) above.

5.1.3 Method for calculating a SEER for units with two speed compressors or two compressors, or cylinder unloading. The calculation procedure described in this section shall be based on the performance of test A and B at each of the compressor speeds for two-speed compressor units, subject to the conditions on condenser fan speed described in 3.1.3.

Units operating with two compressors shall have the SEER calculated in the same manner as two-speed compressor units. The superscripted index $k=1$ (and the term "low-speed") designates the compressor that normally operates at an outdoor dry-bulb temperature of 82° F and $k=2$ (and the term "high speed") denotes operation with both compressors.

In order to evaluate the steady-state capacity $Q_{ss}^{k}(T_j)$, and power input, $E_{ss}^{k}(T_j)$, at temperature T_j for each compressor speed, $k=1$, $k=2$, the results of tests A and B from 5.1 shall be used in the following equation:

$$Q_{ss}^{k}(T_j)=Q_{ss}^{k}(95\text{ F})+\frac{Q_{ss}^{k}(82\text{ F})-Q_{ss}^{k}(95\text{ F})}{95-82}[33-(5\times j)]$$

where

$Q_{ss}^{k}(95\text{ F})$ = Steady-state capacity measured from test A as outlined in 2.1.3.

$Q_{ss}^{k}(82\text{ F})$ = Steady-state capacity measured from test B as outlined in 2.1.3.

$$E_{ss}^{k}(T_j)=E_{ss}^{k}(95\text{ F})+\frac{E_{ss}^{k}(82\text{ F})-E_{ss}^{k}(95\text{ F})}{95-82}[33-(5\times j)]$$

when

$E_{ss}^{k}(95\text{ F})$ = Electrical power input measured using test A as outlined in 2.1.3.

$E_{ss}^{k}(82\text{ F})$ = Electrical power input measured from using test A as outlined in 2.1.3.

The building cooling load $BL(T_j)$ for the four cases described in section 5.1.3.1 through 5.1.3.4 shall be obtained from the following equation:

$$BL(T_j)=\frac{(5\times j)-3}{95-65}\times\frac{Q_{ss}^{k=2}(95\text{ F})}{1.1}$$

where

$Q_{ss}^{k=2}(95\text{ F})$ = Steady-state capacity measured from test A in 2.9 at the high compressor speed.

The value of the degradation coefficient $C_D^{k=1}$ for low compressor speed cycling and $C_D^{k=2}$ for high speed on/off compressor cycling is determined as described in section 2.1.3, or as calculated above in equation (1).

5.1.3.1 Units operating at low compressor speed ($k=1$) for which the steady-state cooling capacity, $Q_{ss}^{k=1}(T_j)$, is greater than or equal to the building cooling load, $BL(T_j)$, evaluate the following equations:

$$(1)\qquad X^{k=1}=\frac{BL(T_j)}{Q_{ss}^{k=1}(T_j)}$$

where

$X^{k=1}$ = Load factor.

$BL(T_j)$ = Building cooling load (Btu/hr) at temperature (T_j) from section 5.1.3.

$Q_{ss}^{k=1}(T_j)$ = Steady-state cooling capacity (Btu/hr) at temperatures (T_j) from section 5.1.3.

$$(2)\qquad \frac{Q(T_j)}{N}=X^{k=1}\times Q_{ss}^{k=1}(T_j)\times\frac{n_j}{N}$$

where

$\frac{Q(T_j)}{N}$ = ratio of total cooling (Btu) in temperature bin j to the number of temperature bin hours.

$\frac{n_j}{N}$ is the fractional number of hours in temperature bin j from 6.1.2.

$$(3)\qquad \frac{E(T_j)}{N}=\frac{X^{k=1}\times E_{ss}^{k=1}(T_j)}{PLF^{k=1}}\times\frac{n_j}{N}$$

where

$\frac{E(T_j)}{N}$ = ratio of Energy usage (watt-hr.) in temperature bin j to the number of temperature bin hours.

$$PLF^{k=1}=1-C_D^{k=1}(1-X^{k=1}).$$

Where C_D the degradation coefficient as described in section 2.1.3 or as calculated above in equation (1).

5.1.3.2 When a unit must alternate between high ($k=2$) and low ($k=1$) compressor speeds to satisfy the building cooling load at a temperature T_j, evaluate the following equations:

$$(1)\qquad X^{k=1}=\frac{Q_{ss}^{k=2}(T_j)-BL(T_j)}{Q_{ss}^{k=2}(T_j)-Q_{ss}^{k=1}(T_j)}$$

$$(2)\qquad X^{k=2}=1-X^{k=1}$$

$$(3)\qquad \frac{Q(T_j)}{N}=[X^{k=1}\times Q_{ss}^{k=1}(T_j)+X^{k=2}\times Q_{ss}^{k=2}(T_j)]\times\frac{n_j}{N}$$

$$(4)\qquad \frac{E(T_j)}{N}=[X^{k=1}\times E_{ss}^{k=1}(T_j)+X^{k=2}\times E_{ss}^{k=2}(T_j)]\times\frac{n_j}{N}$$

5.1.3.3 When a unit must cycle on and off at high compressor speed ($k=2$) in order to satisfy the building cooling load at a temperature T_j, evaluate the equations provided in section 5.1.3.1 replacing ($k=1$) data with the ($k=2$) data.

5.1.3.4 When a unit operates continuously at high compressor speed ($k=2$) at an outdoor temperature T_j evaluate the following equations:

$$(1)\qquad \frac{Q(T_j)}{N}=Q_{ss}^{k=2}(T_j)\times\frac{n_j}{N}$$

$$(2)\qquad \frac{E(T_j)}{N}=E_{ss}^{k=2}(T_j)\times\frac{n_j}{N}$$

5.1.3.5 Calculate the SEER in Btu's/watt-hr. using the values for the terms

$$\frac{Q(T_j)}{N}$$

and

$$\frac{E(T_j)}{N}$$

as determined at each temperature bin according to the applicable conditions described in sections 5.1.3.1 through 5.1.3.4 as follows:

$$SEER=\frac{\sum_{j=1}^{8}\frac{Q(T_j)}{N}}{\sum_{j=1}^{8}\frac{E(T_j)}{N}}$$

5.1.4 Method for calculating a SEER for units with two speed compressor, two compressor or cylinder unloading capable of varying the sensible to total capacity (S/T) ratio. Multi-speed compressor and two-speed compressor units capable of varying the sensible to total capacity ratio (S/T) shall have the seasonal energy efficiency ratio determined as described in section 5.1.3. For such units, the mode of operation selected to determine the steady-state capacities $Q_{ss}{}^{k}(95)$, $Q_{ss}{}^{k}(82)$, $E_{ss}{}^{k}(95)$, $E_{ss}{}^{k}(82)$, and power inputs at each compressor speed $k=1$, $k=2$, for tests A and B is outlined in section 2.10.

5.2 *Calculation of Heating Seasonal Performance Factors (HSPF) for Air-Source Units.*

The testing data and results required to calculate the heating seasonal performance factor (HSPF), in Btu's per watt-hr, shall include the following:

(i) Heating capacities (Btu/hr) from the indoor air enthalpy method for the High Temperature Tests, and the total heating done (Btu's) for the cyclic and frost accumulation tests.
- $\dot{Q}_{ss}(47)$ or $\dot{Q}_{ss}(62)$.
- $\dot{Q}_{ss}(17)$.
- $Q_{cyc}(47)$.
- $Q_{DEF}(35)$.

(ii) Electrical power input to all components (watts) for the steady state tests, and the electrical usage (watt-hours) for the cyclic and frost accumulation tests
- $\dot{E}_{ss}(47)$ or $\dot{E}_{ss}(62)$.
- $\dot{E}_{ss}(17)$.
- $E_{cyc}(47)$.
- $E_{DEF}(35)$.

(iii) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water).

(iv) Air temperature (°F)
- Outdoor dry bulb
- Outdoor wet bulb or dew point
- Indoor dry bulb and
- Indoor wet bulb.

(v) Data as specified in Table II of ASHRAE Standard 37-78.

Where the heating capacities $\dot{Q}_{ss}(47)$, $\dot{Q}_{ss}(62)$ and $\dot{Q}_{ss}(17)$ and the indoor air flow rate are calculated using the equations specified in section 3.8.1 and 7.4 of ASHRAE standard 37-78. The total heating done, $Q_{cyc}(47)$ and $Q_{(DEF}35)$ are calculated using the equations below.

Units not having an indoor fan as part of the model tested shall add 1250 Btu/hr per 1,000 SCFM of indoor air handled to the measured capacity to obtain the total heating capacity, $\dot{Q}_{ss}(17)$, $\dot{Q}_{ss}(47)$ or $\dot{Q}_{ss}(62)$, and add 365 watts per 1,000 SCFM of indoor air handled to the measured power to obtain the total power input, $\dot{E}_{ss}(17)$, $\dot{E}_{ss}(47)$, or $\dot{E}_{ss}(62)$, to the unit.

The coefficients of performance (COP) for the High Temperature Tests $COP_{ss}(62)$ or $COP_{ss}(47)$, and Low Temperature Test, $COP_{ss}(17)$, are calculated as the ratio of the heating capacity in Btu/hr to the product of 3.413 and the power inputs to the indoor fan in watts and the power inputs to the remaining equipment components (including all controls) in watts.

Units which do not have indoor air circulating fans furnished as part of the model shall have their total heating done ($Q_{cyc}(47)$) and energy used $E_{cyc}(47)$ in one complete cycle adjusted for the effect of circulating indoor air equipment power. For units tested without an indoor fan as part of the model, $Q_{cyc}(47)$ shall be increased by a quantity of heat equal to the product of 1250 Btu/hr per 1,000 SCFM, the length of the on-period of the test cycle in hours, and the flow rate of indoor air circulated in units of 1,000 SCFM. The total energy usage, $E_{cyc}(47)$, shall be the sum of the energy usage required for air circulation during the test cycle and the energy used by the remaining equipment components (including all controls) during the test cycle. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 365 watts per 1000 SCFM, the length of the on-period of the test cycle in hours, and the rate of indoor air circulated in units of 1000 SCFM.

The cyclic coefficient of performance, $COP_{cyc}(47)$ is calculated as the ratio of the total heating done ($Q_{cyc}(47)$) in Btu's to the product of 3.413 Btu/watt-hour and the total energy usage ($E_{cyc}(47)$) in watt hours.

The net heating capacity, $\dot{Q}_{DEF}(35)$ (Btu/hr), is the total net heating done over the test period (including any credit for the indoor fan heat) divided by the total length of the test period, in hours. For units with "demand defrost control system", the net heating capacity, $\dot{Q}_{DEF}(35)$, shall be multiplied by an enhancement factor of 1.07 to compensate for improved performance not measured in the Frost Accumulation Test. For units tested without indoor fans, the value determined for $\dot{Q}_{DEF}(35)$ below shall be increased by a quantity of heat equal to the product of 1250 Btu/hr per 1000 SCFM, the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The total energy usage, $E_{DEF}(35)$ shall be the sum of the energy usage required for indoor-air circulation during the test period and the energy used by the remaining equipment components during the test period. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 365 watts per 1000 SCFM, the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The actual heating done during the Cyclic Test, $Q_{cyc}(47)$, shall be determined using the following equation:

$$(1)\qquad Q_{cyc}(47)=\frac{60\times\dot{V}\times C_{pa}\times\Gamma}{[V_n{}^1\times(1+W_n)]}$$

where

$\dot{V}$ = the flow rate during the on-period calculated in accordance with section 7.4 of ASHRAE Standard 37-78 in CFM.
C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air, (Btu/lb-°F).
$V_n{}^1$ = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).
W_n = Humidity ratio (lb/lb).

and Γ (hr-°F), which is described by the equation:

$$\Gamma=\int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}}[T_{a2}(t)-T_{a1}(t)]dt$$

where

$T_{a1}(t)$ = Dry-bulb temperature of air entering the indoor coil (°F) at time (t).
$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (°F) at time (t).

The net heating, Q_{DEF} (35) in Btu's done during the test period shall be determined using the following equation:

$$(2)\qquad Q_{DEF}(35)=\frac{60\times\dot{V}\times C_{pa}\times\Gamma}{[V_n{}^1\times(1+W_n)]}$$

where

$\dot{V}$ = the average of the air flow rate calculated at four or more time intervals throughout the heating phase of the test using the equation in section 7.4 of ASHRAE Standard 37-78.
C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air, (Btu/lb-°F).
$V_n{}^1$ = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).
W_n = Humidity ratio (lb/lb).

and Γ (hr.-°F), which is described by the equation:

$$\Gamma=\int_{\text{(time of defrost termination}}^{\text{(time of next defrost termination)}}\times[T_{a2}(t)-T_{a1}(t)]dt$$

where

$T_a{}^1(t)$ = Dry-bulb temperature of air entering the indoor coil (°F) at time (t).
$T_a{}^2(t)$ = Dry-bulb temperature of air leaving the indoor coil (°F) at time (t).

The cyclic degradation coefficient shall be calculated as follows:

$$(3)\qquad C_D=\frac{1-\frac{COP_{cyc}(47)}{COP_{ss}(47)}}{1-HLF}$$

where

C_D = the cyclic degradation coefficient rounded to the nearest .01
$COP_{cyc}(47)$ as defined above
$COP_{ss}(47)$ as defined above
HLF is the heating load factor calculated as follows:

$$HLF=\frac{Q_{cyc}(47)}{\dot{Q}_{ss}(47)\times\tau}$$

where

$Q_{cyc}(47)$ as defined above

$\dot{Q}_{ss}(47)$ as defined above

τ = Duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

5.2.1 Calculation of the heating seasonal performance factor (HSPF) for air-source heat pumps with single speed compressors.

For each climatic region listed in section 6.2.4, and for design heating requirements equal to both the standardized minimum and maximum design heating requirements defined below, calculate the HSPF defined as:

$$HSPF = \frac{\sum_j \frac{n_j}{N} BL(T_j)}{\left[\sum_j \frac{n_j}{N} \frac{X(T_j)}{PLF(X)} \delta(T_j)\dot{E}(T_j)(T_j) + \sum_j \frac{RH(T_j)}{N}\right]},$$

where

j = 1,2,3, . . . , n corresponds to the j^{th} temperature bin.

n = total number of non-zero temperature bins in the climatic region.

T_j = (67−5_j) is the representative temperature of the j^{th} bin, (°F).

$\sum_j$ = indicated the quantity following the symbol is to be summed over all temperature bins.

$\frac{RH(T_j)}{N}$ = supplementary resistance heat term at temperature T_j required in those cases where the heat pump automatically turns off ($T_j < T_{off}$) or when it is needed to meet the balance of the building heating requirements. (watts).

$\frac{n_j}{N}$ = is the number of hours in the j^{th} temperature bin divided by $N \equiv \sum_j n_j$ and is referred to as the "fractional hours in j^{th} temperature bin".

3.413 = is a conversion factor which converts watt hours to Btu.

$BL(T_j)$ = building load at temperature T_j, (Btu/hr).

$\delta(T_j)$ = heat pump low temperature cut-out factor.

$X(T_j)$ = heat pump heating load factor.

$PLF(X)$ = heat pump part load factor.

The quantities $BLCT_j$, $\delta(T_j)$, $X(T_j)$, $PLF(X)$ and

$$\frac{RH(T_j)}{N}$$

are defined by the following equations:

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right)(C)(DHR)$$

where

(C) = 0.77 is a correction factor which tends to improve the agreement between calculated and measured building loads

(DHR) = the minimum and maximum design heating requirement which the heat pump is likely to encounter when installed in a residence, rounded off to the nearest standardized DHR in section 6.2.6 in Btu/hr

where

(minimum design heating requirement)

$$= \begin{cases} \dot{Q}_{ss}(47)\ \frac{(65 - T_{OD})}{60}, & \text{for regions I,} \\ & \text{II, III, IV, and VI} \\ \dot{Q}_{ss}(47), & \text{for region V} \end{cases}$$

and

maximum design heating requirement

$$= \begin{cases} 2\dot{Q}_{ss}(47)\ \frac{(65 - T_{OD})}{60}, & \text{for regions I,} \\ & \text{II, III, IV, and VI} \\ 2.2\dot{Q}_{ss}(47), & \text{for region V} \end{cases}$$

where

$\dot{Q}_{ss}(47)$ is the heat pump capacity measured during the High Temperature Test @ 47° F

T_{OD} is the outdoor design temperature given in section 6.2.4

$$\delta(T_j) = \begin{cases} 0;\ T_j \le T_{OFF} \\ \quad \text{or } \frac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} < 1 \\ \frac{1}{2};\ T_{OFF} < T_j \le T_{ON} \\ \quad \text{and } \frac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \ge 1 \\ 1;\ T_j > T_{ON} \\ \quad \text{and } \frac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \ge 1 \end{cases}$$

$$X(T_j) = \begin{cases} \frac{BL(T_j)}{\dot{Q}(T_j)};\ \dot{Q}(T_j) \ge BL(T_j) \\ 1;\ \dot{Q}(T_j) \le BL(T_j) \end{cases}$$

$$PLF(X) = 1 - C_D(1 - X(Tj))$$

$$\frac{RH(T_j)}{N} = \frac{[BL(T_j) - \dot{Q}(T_j)X(T_j)\delta(T_j)]\frac{n_j}{N}}{3.413}$$

where

T_{OFF} = the outdoor temperature that the compressor is automatically shut off at (If no such temperature exists, T_j is always greater than T_{OFF} and T_{ON}.)

T_{ON} = the outdoor temperature that the compressor is automatically turned on (if applicable) if designed for low temperature automatic shut-off.

C_D = degradation factor determined described in section 5.2 and 2.2.1.

In using the above equation to calculate HSPF, the heat pump capacity in Btu/hr, $\dot{Q}$, and the power in watts, $\dot{E}$, shall be obtained as follows:

$$\dot{Q}(T_j) = \begin{cases} \dot{Q}_{ss}(17) + \frac{(\dot{Q}_{ss}(47) - \dot{Q}_{ss}(17)) \times (T_j - 17)}{30}, \\ \quad T_j \ge 45°F \text{ or } T_j \le 17°F \\ \dot{Q}_{ss}(17) + \frac{(\dot{Q}_{DEF}(35) - \dot{Q}_{ss}(17)) \times (T_j - 17)}{18}, \\ \quad 17°F < T_j < 45°F \end{cases}$$

$$\dot{E}(T_j) = \begin{cases} \dot{E}_{ss}(17) + \frac{(\dot{E}_{ss}(47) - \dot{E}_{ss}(17)) \times (T_j - 17)}{30}, \\ \quad T_j \ge 45°F \text{ or } T_j \le 17°F \\ \dot{E}_{ss}(17) + \frac{(\dot{E}_{DEF}(35) - \dot{E}_{ss}(17)) \times (T_j - 17)}{18}, \\ \quad 17°F < T_j < 45°F \end{cases}$$

where $\dot{Q}_{ss}(47)$ and $\dot{E}_{ss}(47)$ and $\dot{Q}_{DEF}(35)$ and $\dot{E}_{ss}(47)$ and $\dot{Q}_{ss}(17)$ and $\dot{E}_{ss}(17)$ are the capacities (in Btu/hr) and powers (in watts), measured during the High Temperature Test, the Frost Accumulation test, and the Low Temperature Test, respectively.

Once the maximum and minimum HSPF and operating cost values have been obtained for each region, the HSPF and operating cost shall be determined for each standardized design heating requirement (see section 6.2.6) between the maximum and minimum design heating requirements by means of interpolation.

5.2.2 Calculation of the heating seasonal performance factor (HSPF) for air source heat pumps with a two-speed compressor, two compressors, or cylinder unloading.

For each climatic region listed in section 6.2.4, and for design heating requirements equal to both the standardized minimum and maximum design heating requirements defined below, calculate the HSPF defined as:

$$HSPF = \frac{\sum_j \frac{n_j}{N} BL(T_j)}{\left[\sum_j \frac{E(T_j)}{N} + \sum_j \frac{RH(T_j)}{N}\right]}$$

where

$\sum_j$ as defined in 5.2.1

$\frac{n_j}{N}$ as defined in 5.2.1

T_j as defined in 5.2.1

and $BL(T_j)$ is the building load at temperature T_j, in Btu/hr, calculated by:

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right) \times (C) \times (DHR)$$

where

(C) = 0.77 is a correction factor which tends to improve the agreement between calculated and measured building loads

(DHR) = the minimum and maximum design heating requirement which the heat pump is likely to encounter when installed in a residence, rounded off to the nearest standardized DHR in section 6.2.6 in Btu's per hour

where

(minimum design heating requirement)

$$= \begin{cases} \dot{Q}_{ss}^{(k=2)}(47)\ \frac{(65 - T_{OD})}{60}, & \text{for regions I,} \\ & \text{II, III, IV, and VI} \\ \dot{Q}_{ss}^{(k=2)}(47), & \text{for region V} \end{cases}$$

and

(maximum design heating requirement)

$$= \begin{cases} 2\dot{Q}_{ss}^{(k=2)}(47)\ \frac{(65 - T_{OD})}{60}, & \text{for regions I,} \\ & \text{II, III, IV and VI} \\ 2.2\dot{Q}_{ss}^{(k=2)}(47) & \text{for region V} \end{cases}$$

where

$\dot{Q}_{ss}^{(k=2)}(47)$ is the heat pump capacity measured during the high temperature performance test at 47°F, with the unit operating at the high compressor speed or with both compressors in operation, in Btu/hr

T_{OD} is the outdoor design temperature given in section 6.2.4 in degrees °F.

NOTE: The superscript ($k=1$) and ($k=2$) refer to the heat pump operating at low speed or single compressor operation and high speed or two compressor operation respectively.

$\frac{E(T_j)}{N}$ is the heat pump electrical energy usage in the j^{th} temperature bin divided by the total number of bin hours and is evaluated according to the four possible cases of heat pump operation denoted below in watts.

$\frac{RH(T_j)}{N}$ as defined in 5.2.1 and is evaluated according to the four possible cases of heat pump operation denoted below (in watts).

Case I.—Units operating at low compressor speed or with a single compressor, i.e., $k=1$, for which the building heating load, $BL(T_j)$ is less than or equal to the heating capacity, $\dot{Q}^{k=1}(T_j)$.

$$BL(T_j) \leq \dot{Q}^{k=1}(T_j)$$

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=1}(T_j) X^{k=1}(T_j) \delta'(T_j)}{PLF^{k=1}} \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N} BL(T_j)[1-\delta'(T_j)]}{3.413}$$

$$X^{k=1}(T_j) = \frac{BL(T_j)}{\dot{Q}^{k=1}(T_j)}$$

$$PLF^{k=1} = 1 - C_D^{k=1}(1 - X_j^{k=1})$$

$$\delta'(T_j) = \begin{cases} 0; & T_j \leq T_{OFF} \\ \frac{1}{2}; & T_{OFF} < T_j \leq T_{ON} \\ 1; & T_j > T_{ON} \end{cases}$$

Case II.—Units alternating between high speed or two compressor operation ($k=2$) and low speed or single compressor operation ($k=1$) to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j) < BL(T_j) < \dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N} = [\dot{E}^{k=1}(T_j) X^{k=1}(T_j) + \dot{E}^{k=2}(T_j) X^{k=2}(T_j)] \delta'(T_j) \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N} BL(T_j)[1-\delta'(T_j)]}{3.413}$$

$$X^{k=1}(T_j) = \frac{\dot{Q}^{k=2}(T_j) - BL(T_j)}{\dot{Q}^{k=2}(T_j) - \dot{Q}^{k=1}(T_j)}$$

$$X^{k=2}(T_j) = 1 - X^{k=1}(T_j)$$

$$\delta'(T_j) \begin{cases} 0; & T_j \leq T_{OFF} \\ \frac{1}{2}; & T_{OFF} < T_j \leq T_{ON} \\ 1; & T_j > T_{ON} \end{cases}$$

Case III.—Units cycling on and off at high compressor speed or cycling both compressors on and off together ($k=2$) in order to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j) < BL(T_j) < \dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=2}(T_j) X^{k=2}(T_j) \delta''(T_j) \frac{n_j}{N}}{PLF^{k=2}}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N} BL(T_j)[1-\delta''(T_j)]}{3.413}$$

$$X^{k=2}(T_j) = \frac{BL(T_j)}{\dot{Q}^{k=2}(T_j)}$$

$$PLF^{k=2} = 1 - C_D^{k=2}(1 - X^{k=2}(T_j))$$

$$\delta''(T_j) = \begin{cases} 0; & T_j \leq T_{OFF} \\ \frac{1}{2}; & T_{OFF} < T_j \leq T_{ON} \\ 1; & T_j > T_{ON} \end{cases}$$

Case IV.—Units operating continuously at high compressor speed or with both compressors in continuous operation ($k=2$) in order to satisfy the building heating load at temperature T_j.

$$BL(T_j) \geq \dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N} = \dot{E}^{k=2}(T_j) X^{k=2}(T_j) \delta''(T_j) \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{[BL(T_j) - \dot{Q}^{k=2}(T_j) X^{k=2}(T_j) \delta''(T_j)] \frac{n_j}{N}}{3.413}$$

$$X^{k=2}(T_j) = 1.0$$

$$\delta''(T_j) \begin{cases} 0; \; T_j \leq T_{OFF} \text{ or } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} < 1 \\ \frac{1}{2}; \; T_{OFF} < T_j \leq T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} \geq 1 \\ 1; \; T_j \geq T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} \geq 1 \end{cases}$$

Where in each of the above cases

$X(T_j)$ = heat pump heating load factor.
PLF = heat pump part load factor (not required for cases II and IV.)
$\delta''(T_j)$ = heat pump low temperature cut-out factor.
T_{off} as defined in 5.2.1.
T_{on} as defined in 5.2.1.
$C_D^{k=2}$ = the part load degradation factor described in section 2.2.1 and 5.2 for the unit cycling at high compressor speed or with both compressors simultaneously cycling (if applicable).
$C_D^{(k=1)}$ = the part load degradation factor described in section 2.2.1 and 5.2 for the unit cycling at low compressor speed or with the single compressor that normally operates at low heating loads.

$$\dot{Q}^{k=1}(T_j) = \begin{cases} \dot{Q}_{ss}^{k=1}(47) \\ \quad + \dfrac{(\dot{Q}_{ss}^{k=1}(62) - \dot{Q}_{ss}^{k=1}(47))}{15} \times (T_j - 47); \\ \qquad T_j \geq 40° \text{ F} \\ \dot{Q}_{ss}^{k=1}(17) \\ \quad + \dfrac{(\dot{Q}_{DEF}^{k=1}(35) - \dot{Q}_{ss}^{k=1}(17))}{18} \times (T_j - 17); \\ \qquad 17° \text{ F} \leq T_j < 40° \text{ F} \\ \dot{Q}_{ss}^{k=1}(17) \\ \quad + \dfrac{(\dot{Q}_{ss}^{k=1}(47) - \dot{Q}_{ss}^{k=1}(17))}{30} \times (T_j - 17); \\ \qquad T_j < 17° \text{ F} \end{cases}$$

$$\dot{Q}^{k=2}(T_j) = \begin{cases} \dot{Q}_{ss}^{k=2}(17) \\ \quad + \dfrac{(\dot{Q}_{ss}^{k=2}(47) - \dot{Q}_{ss}^{k=2}(17))}{30} \times (T_j - 17); \\ \qquad T_j \geq 45° \text{ F or } T_j \leq 17° \text{ F} \\ \dot{Q}_{ss}^{k=2}(17) \\ \quad + \dfrac{(\dot{Q}_{DEF}^{k=2}(35) - \dot{Q}_{ss}^{k=2}(17))}{18} \times (T_j - 17); \\ \qquad 17° \text{ F} < T_j < 45° \text{ F} \end{cases}$$

$$\dot{E}^{k=1}(T_j) = \begin{cases} \dot{E}_{ss}^{k=1}(47) \\ \quad + \dfrac{(\dot{E}_{ss}^{k=1}(62) - \dot{E}_{ss}^{k=1}(47))}{15} \times (T_j - 47); \\ \qquad T_j \geq 40° \text{ F} \\ \dot{E}_{ss}^{k=1}(17) \\ \quad + \dfrac{(\dot{E}_{DEF}^{k=1}(35) - \dot{E}_{ss}^{k=1}(17))}{18} \times (T_j - 17); \\ \qquad 17° \text{ F} \leq T_j < 40° \text{ F} \\ \dot{E}_{ss}^{k=1}(17) \\ \quad + \dfrac{(\dot{E}_{ss}^{k=1}(47) - \dot{E}_{ss}^{k=1}(17))}{30} \times (T_j - 17); \\ \qquad T_j < 17° \text{ F} \end{cases}$$

$$\dot{E}^{k=2}(T_j) = \begin{cases} \dot{E}_{ss}^{k=2}(17) \\ \quad + \dfrac{(\dot{E}_{ss}^{k=2}(47) - \dot{E}_{ss}^{k=2}(17))}{30} \times (T_j - 17); \\ \qquad T_j \geq 45° \text{ F or } T_j \leq 17° \text{ F} \\ \dot{E}_{ss}^{k=2}(17) \\ \quad + \dfrac{(\dot{E}_{DEF}^{k=2}(35) - \dot{E}_{ss}^{k=2}(17))}{18} \times (T_j - 17); \\ \qquad 17° \text{ F} < T_j < 45° \text{ F} \end{cases}$$

For each of the six regions specified in section 6.2.5, calculate the heating seasonal performance factors and seasonal operating costs corresponding to the standardized maximum and minimum design heating requirements and for all other standardized design heating requirements (see section 6.2.6) between the maximum and the minimum.

5.3 *Calculations of the Actual Representative Regional Annual Performance Factors for Air Source Central Air Conditioners (Heat Pumps) Which Provide Both Heating and Cooling.*

5.3.1 Calculation of actual regional annual performance factors (APF_A) for a particular location and for each standardized design heating requirement.

$$(APF_A)=\frac{(CLH_A)(\dot{Q}_{ss}{}^k(95F))+(HLH_A)(DHR)(C)}{\frac{(CLH_A)(\dot{Q}_{ss}{}^k(95F))}{(SEER)}+\frac{(HLH_A)(DHR)(C)}{(HSPF)}}$$

where

- (CLH_A) is the actual cooling load hours for the particular location, determined from the map in section 6.1.3.
- $(\dot{Q}_{ss}{}^k(95F))$ is defined in 5.1.
- (DHR) is defined in 5.2.2.
- (C) is defined in 5.2.2.
- (HLH_A) is the actual heating load hours for the particular location determined from the map in section 6.2.5.
- $(SEER)$ is the seasonal energy efficiency ratio determined in section 5.1.
- $(HSPF)$ is the heating seasonal performance factors as determined in section 5.2 for each standardized design heating requirement within the particular location's region or for the actual design heating requirement if known.

where the particular location's region is determined from the map in section 6.2.5 and, the standardized design heating requirements within the region are determined in sections 5.2 and 6.2.6.

5.3.2 Calculation of representative regional annual performance factors (APF_R) for each region and for each standardized design heating requirement.

$$(APF_R)=\frac{(CLH_R)(\dot{Q}_{ss}{}^k(95F))+(HLH_R)(DHR)(C)}{\frac{(CLH_R)(\dot{Q}_{ss}{}^k(95F))}{SEER}+\frac{(HLH_R)(DHR)(C)}{HSPF}}$$

- $(\dot{Q}_{ss}{}^k(95F))$ is defined in 5.1.
- (DHR) is defined in 5.2.2.
- (C) is defined in 5.2.2.

where

- (CLH_R) is the representative cooling load hours for each heating load hours region, as determined in section 6.3.
- (HLH_R) is the representative heating load hours for each region as determined in section 6.2.5.
- $(SEER)$ is the seasonal energy efficiency ratio as determined in section 5.1.
- $(HSPF)$ is the heating seasonal performance factor as determined in section 5.2 for each region and for each standardized design heating requirement within each region.

where the regions are listed in section 6.2.5 and, the standardized design heating requirements within the regions are determined in sections 5.2 and 6.2.6.

5.4 *Calculations of Seasonal Energy Efficiency Ratio for Water Source Units Which Provide Both Heating and Cooling.* (Reserved)

5.5 *Calculation of Heating Seasonal Performance Factor for Water Source Units Which Provide Both Heating and Cooling.* (Reserved)

5.6 *Calculation of Annual Performance Factor For Water Source Units Which Provide Both Heating and Cooling.* (Reserved)

BILLING CODE 6450-01-M

6.0 Reference Material

6.1 Cooling reference material

6.1.1 -Test operating and test condition tolerance for cyclic dry-coil tests

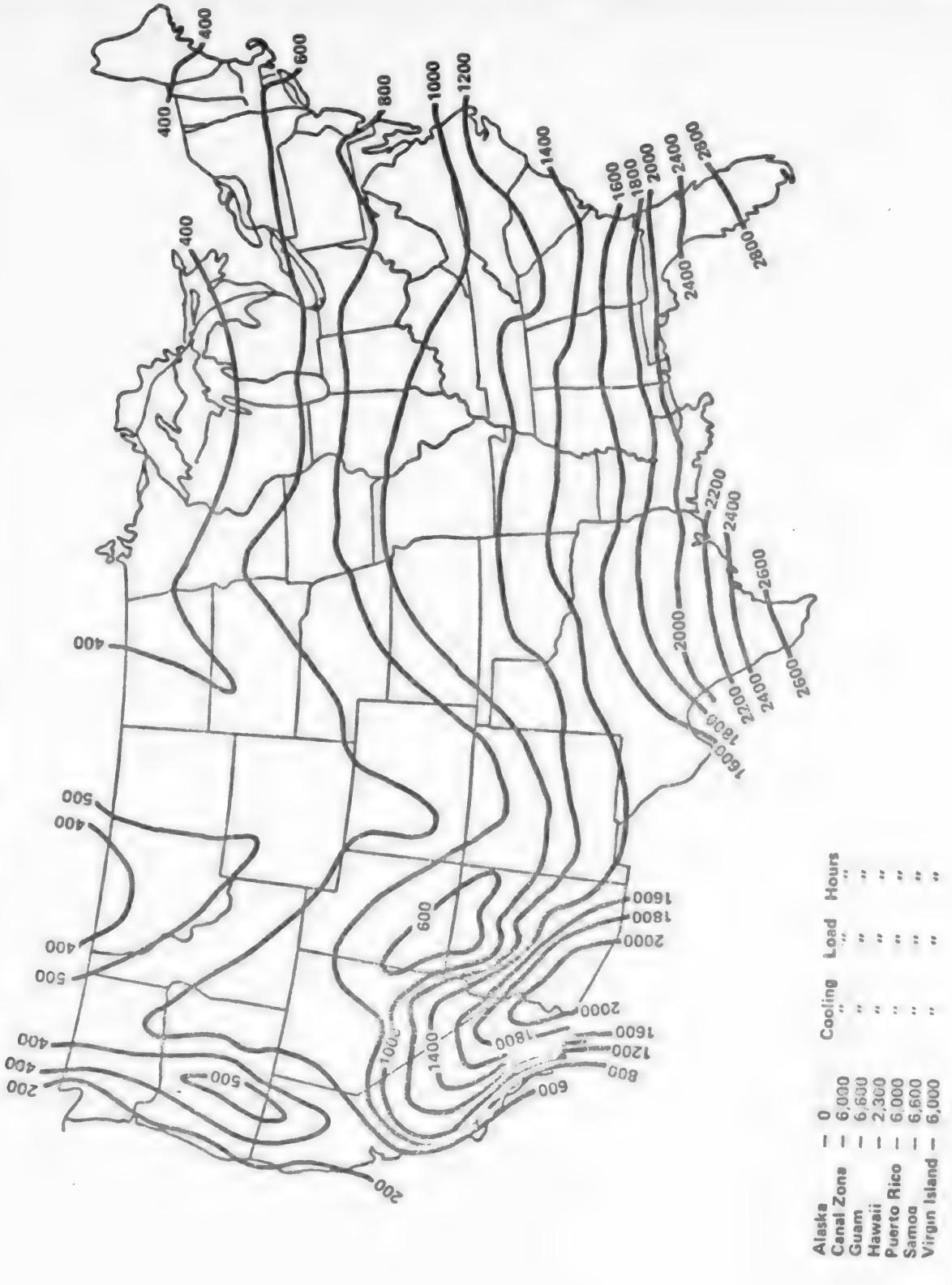
Readings, remarks	Test operating tolerance (total observed range)	Test condition tolerance (variation of average from specified test condition)
Outdoor dry-bulb air temperature, Fahrenheit: Entering	2.0	0.5
Indoor dry-bulb air temperature, Fahrenheit: Entering	2.0	0.5
Indoor wet-bulb air temperature, Fahrenheit: Entering	(*)	(*)
After the 1st 30 Sec after compressor startup. External resistance to airflow, inches water	0.05	0.02
Nozzle pressure drops, percent of reading	2.0	.....
Electrical voltage inputs to the test unit, percent.	2.0	.....

* Shall at no time exceed that value of the wet-bulb temperature which results in the production of condensate by the indoor coil at the dry-bulb temperature existing for the air entering the indoor portion of the unit.

6.1.2 -Distribution of fractional hours in temperature bins to be used for calculation of the SEER for 2-speed compressor and 2-compressor units

Bin temperature range (degrees Fahrenheit)	Representative temperature bin for (degrees Fahrenheit)	Fraction of total temperature bin hours n_j/N
Bin No., j:		
1..... 65 to 69...	67	.214
2..... 70 to 74...	72	.231
3..... 75 to 79...	77	.216
4..... 80 to 84...	82	.161
5..... 85 to 89...	87	.104
6..... 90 to 94...	92	.052
7..... 95 to 99...	97	.018
8..... 100 to 104.	102	.004

6.1.3 Distribution of Actual Cooling Load Hours (CLH_A) Throughout the United States



6.2 Heating Reference Material

6.2.1 Test operating and test condition tolerance for

Steady-State High Temperature Test [at 47°F (8.3°C) or 62°F (16.7°C)]
and
Low Temperature Test [at 17°F (-8.3°C)]

	Test Operating* Tolerance	Test Condition** Tolerance
Indoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	2.0	---
Indoor wet-bulb, °F		
Entering	1.0	---
Leaving	1.0	---
Outdoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	2.0	---
Outdoor wet-bulb, °F		
Entering	1.0	0.3
Leaving	1.0	---
External resistance to air flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition.

6.2.2 Test operating and test condition tolerances for the on-period portion of cyclic performance tests.

	Test Operating Tolerances* (Applies after the 1st 30 sec. after compressor start-up)	Test Condition Tolerance**
Indoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	---	---
Indoor wet-bulb, °F		
Entering	1.0	---
Leaving	---	---
Outdoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	---	---
Outdoor wet-bulb, °F		
Entering	2.0	1.0
Leaving	---	---
External resistance to air-flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition.

6.2.3 Test operating and test condition tolerances for frost accumulation tests

	Testing Operating Tolerance[1] During Heating	Testing Operating Tolerance[1] During Defrost	Test Condition Tolerance[2] (Heating Portion Only)
Indoor dry-bulb, °F			
Entering	2.0	4.0[3]	0.5
Leaving	---	---	---
Indoor wet-bulb, °F			
Entering	1.0	---	---
Leaving	---	---	---
Outdoor dry-bulb, °F			
Entering	2.0	10.0	1.0
Leaving	---	---	---
Outdoor dew-point, °F			
Entering	1.5	---	0.7
Leaving	---	---	---
External resistance to air-flow, inches of water	.05	---	.02
Electrical voltage, %	2.0	---	---

1 Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value. Test Operating Tolerance During Heating applies when the heat pump is in the heating mode, except for the first 5 minutes after termination of a defrost cycle. Test Operating Tolerance During Defrost applies during a defrost cycle and during the first 5 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.

2 Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition. Test Condition Tolerance applies only when the heat pump is operating in the heating mode.

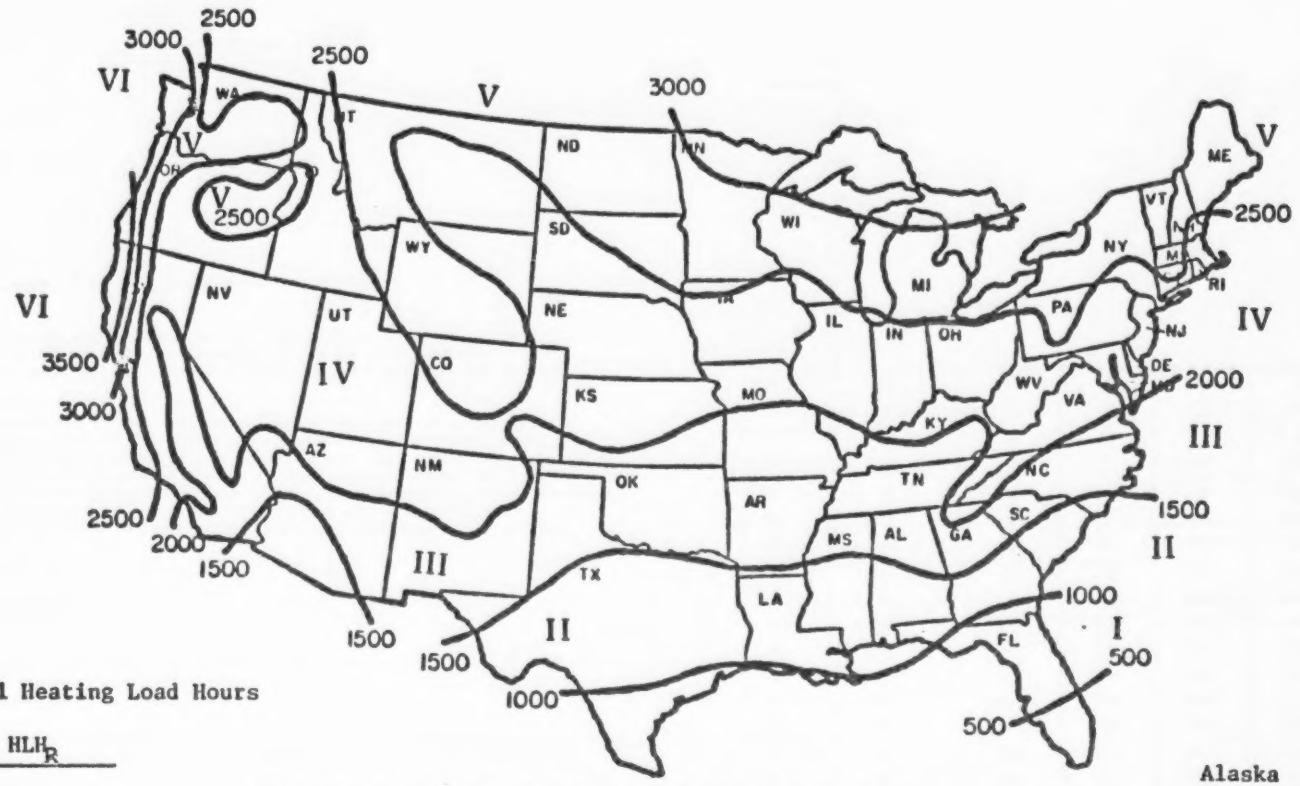
3 Not applicable during defrost if the indoor fan is off.

6.2.4 Distribution of fractional hours in temperature bins, heating load hours and outdoor design temperature for the different climatic regions

Region		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	2750*
Outdoor Design Temperature, T_{OD} for the region		37	27	17	5	-10	30
Fractional Hours:							
Bin #	T_j (°F)						
j = 1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	22	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	-3	0	0	0	.002	.018	0
15	-8	0	0	0	.001	.010	0
16	-13	0	0	0	0	.005	0
17	-18	0	0	0	0	.002	0
18	-23	0	0	0	0	.001	0

* Pacific Coast Region

6.2.5 Actual heating load hours (HLH_A) and regional heating load hours (HLH_R) for the United States



This map is reasonably accurate for most parts of the United States but is necessarily highly generalized, and consequently not too accurate in mountainous regions, particularly in the Rockies.

Regional Heating Load Hours

Region	HLH_R
I	750
II	1250
III	1750
IV	2250
V	2750
VI	2750

Alaska - 3500 HLH
Hawaii and Territories - 0 HLH

6.2.6 Standard Design Heating Requirements (Btu/hr)

5,000	25,000	50,000	90,000
10,000	30,000	60,000	100,000
15,000	35,000	70,000	110,000
20,000	40,000	80,000	130,000

6.3 Representative Cooling Load Hours (CLH_R) for Each Heating Load Hours Region

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

6.4 Ground Water Temperature Map (Reserved)

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